Vol. 17, No. 6 June, 1961

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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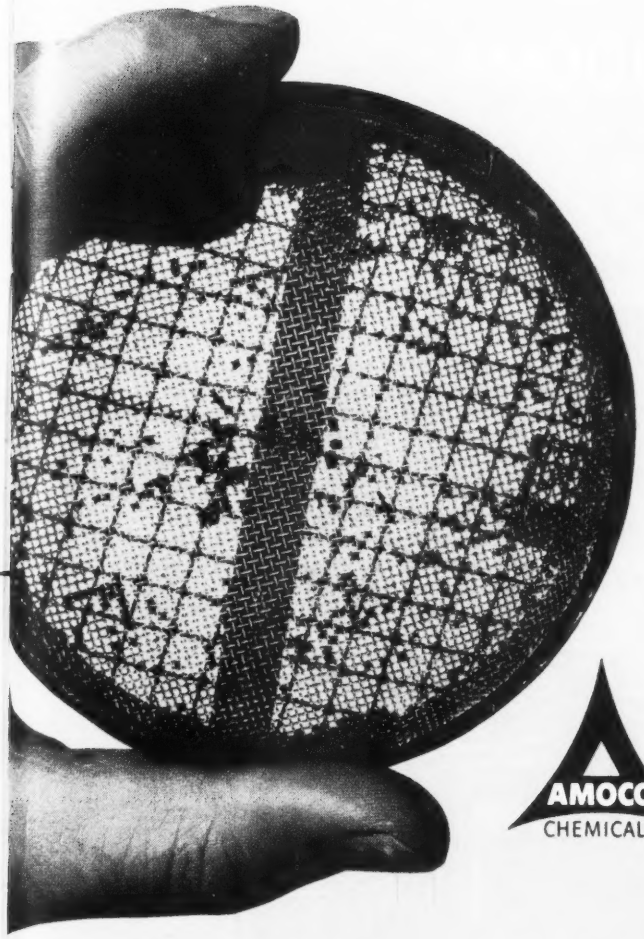

The AMOCO 200 Series Additives are blends of a unique barium detergent and an especially effective zinc dialkyl dithiophosphate inhibitor. Motor oils containing these additives are superior in ability to prevent varnish and sludge deposits in stop-and-go operation. These performance characteristics have been demonstrated in laboratory tests—including Sequence V and other sequences of the MS test procedure —and in millions of miles of field tests.

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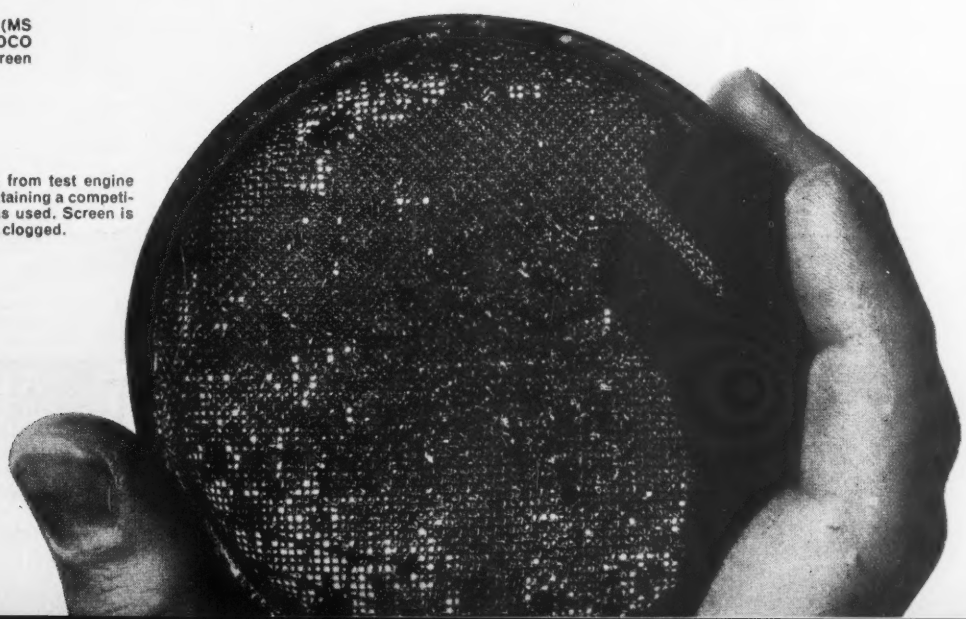
Multi-cylinder laboratory test (MS sequence V). With an AMOCO 200 Series Additive, oil screen (above) shows no clogging.

Screen (below) from test engine in which oil containing a competitive additive was used. Screen is more than 75% clogged.



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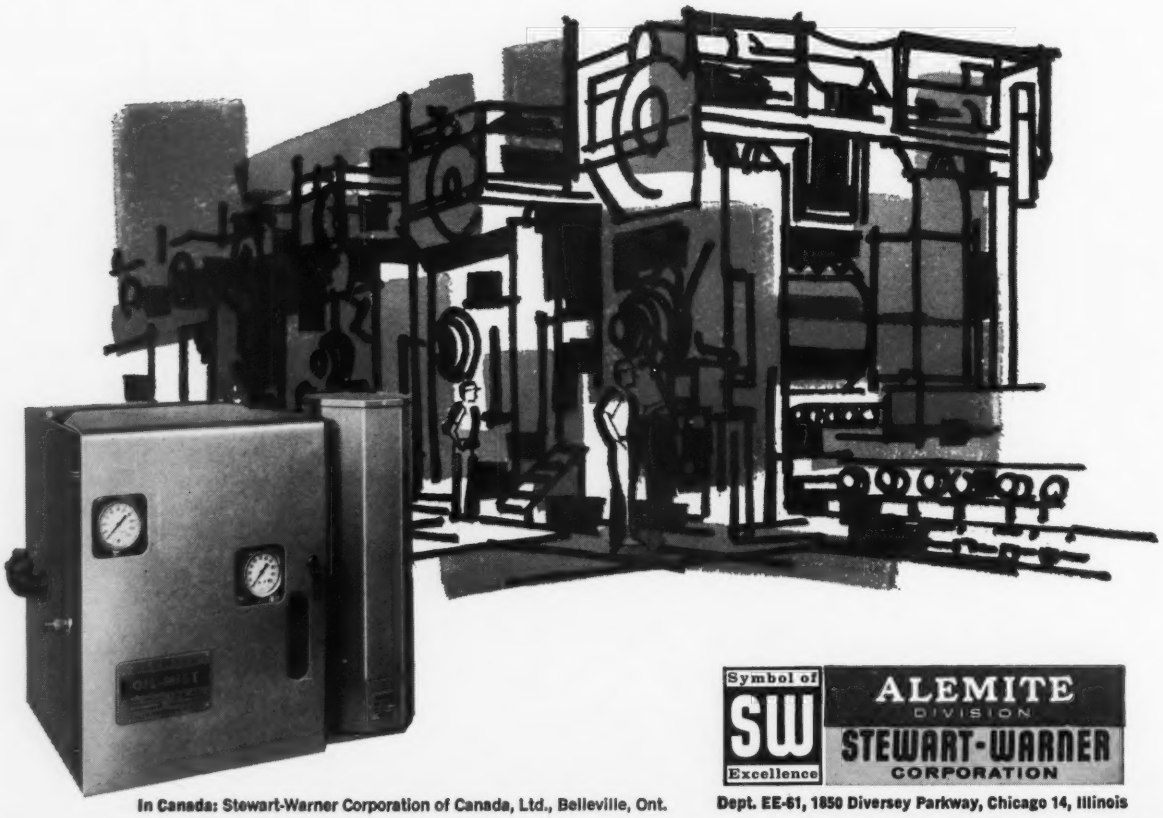
Alemite Oil-Mist with Thermo-Aire will improve mill operating efficiency...lengthen bearing life . . . cut oil consumption and prevent product spoilage.

Write for illustrated brochure form 37-32, or see your Alemite representative for the amazing story of Alemite Oil-Mist with Thermo-Aire and how you can benefit from it!



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Lubrication Engineering

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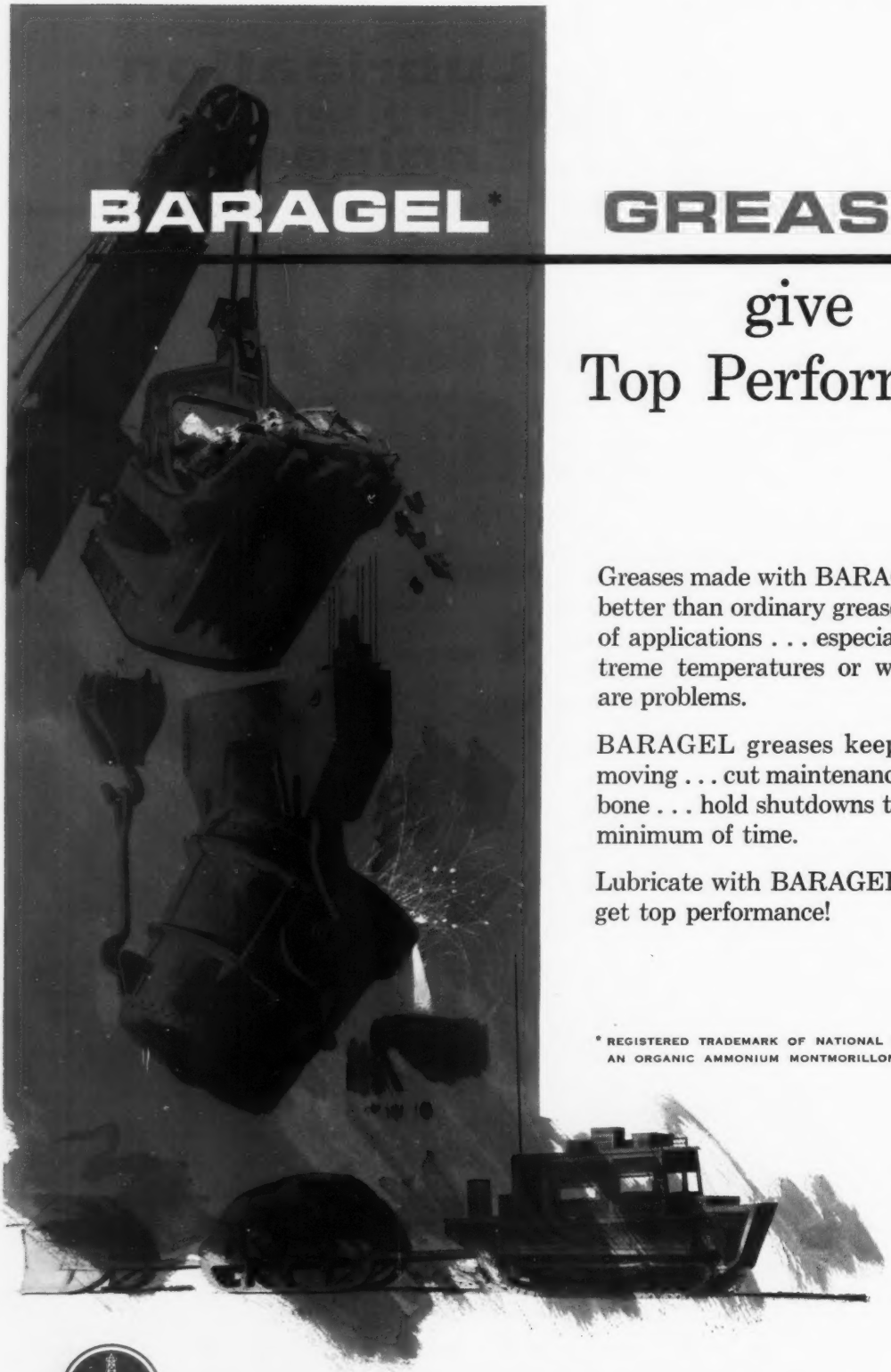

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LUBRICATION IN THE NEWS

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RESEARCH STUDY ON LARGE DIAMETER BEARINGS PUBLISHED

The new Kaydon research study, titled *Feasibility Study of Large Diameter Anti-Friction Bearings for Radar Antenna Applications,* was developed under an Air Force research program. This 227 page research report presents more than 100 design graphs, drawings, and tables to give information on internal load distribution, contact angles, capacity, life, deflection, mean compressive stress, contact eclipse, and other information for specific ball and roller sizes. A 69 item Bibliography is also included. This book is available from Kaydon Engineering Corp., McCracken St., Muskegon, Mich., for $25.00. (Source: Kaydon Engineering Corp.)

DESIGN CRITERIA STUDIED ON ANTI-FRICTION BEARINGS FOR HIGH TEMPERATURE USE

Research to develop design criteria for anti-friction airframe bearings for use in air and space vehicles is described in one of four Air Force reports released to science and industry through the Office of Technical Services, Business and Defense Services Administration, U.S. Department of Commerce, Washington 25, D. C.

Research On Developing Design Criteria For Anti-Friction Airframe Bearings For High Temperature Use. W. F. O'Rourke. General Motors Corporation, for Wright Air Development Division, U.S. Air Force. October 1960. 138 pages. Order PB 171 151 from OTS, U.S. Department of Commerce, Washington 25, D. C., $2.75. A study to develop design criteria for anti-friction airframe bearings for use in air and space vehicles is described. Desired operation includes 40,000 oscillatory cycles at 600 F to 1200 F under a radial load corresponding to approximately 500,000 psi mean compressive stress for a one-inch bore ball bearing. Bearings made from three material combinations were tested; the results, extrapolated to four other sizes, indicated that performance is limited by the required load with a maximum of 26,500 cycles of operation having been obtained at 900 F and one-half the prescribed load, and 40,000 cycles at one-third the prescribed load. The most promising materials were tool steel and stainless steel for rings; titanium carbide for balls for temperatures to 900 F; and chromium carbide for rings and cobalt alloy for balls at 1200 F. None of the 25 lubricating aids showed a significant improvement. (Source: U. S. Department of Commerce, Washington, D. C.)

AEROSPACE BEARINGS OPERATED AT 1000 F.

Experimental bearings of exotic materials and special alloys for possible use in missile, aircraft, and gas turbine engines have run at 1000 F. and above—without normal lubrication—it has been disclosed by SKF Industries' Research Laboratory.

The report stated that it is believed this is the first time this type of bearing has run at this extreme temperature over prolonged periods and with substantial loads and speeds.

"Made of carbides, ceramics, and other specially developed heat-resisting alloys to operate at high temperatures, these rolling contact bearings operated where conventional steel bearings would fail and normal lubricants burn away," said Thomas W. Morrison, director of engineering and research. (Source: SKF Industries)

NEW LABORATORIES' BUILDING FOR THE FRANKLIN INSTITUTE

It has been announced that the Philadelphia Industrial Development Corporation will build a $2,500,000, 6-story research building for The Franklin Institute on a site adjacent to the Institute's main building in Philadelphia, Pa. The Institute will occupy the facility under a long term lease.

When completed in late 1962, the 120,000 square foot building will enable the institute to expand the activities of its research laboratories, which serve both private industry and Federal agencies, including the Defense Department. (Source: The Franklin Institute).

REFRACTORY MATERIAL FRICTION AND WEAR BEHAVIOR EVALUATED

Factors governing the friction and wear behavior of ceramics, cermets, and high-temperature alloys are evaluated in an Air Force research report released to science and industry through the Office of Technical Services, Business and Defense Services Administration, U. S. Department of Commerce, Washington 25, D. C.

Characteristics Governing the Friction and Wear Behavior of Refractory Materials for High-Temperature Seals and Bearings. L. B. Sibley and others, Battelle Memorial Institute for Wright Air Development Division, U. S. Air Force. May 1960. 54 pages. Order PB 171 010 from OTS, U. S. Department of Commerce, Washington 25, D. C., $1.50. A study was made of the basic factors involved in the wear and friction of ceramics, cermets, and high-temperature alloys sliding at speeds of 100 to 200 fps and temperatures from 500 to 1000 C. Bearing pressures covered a range of 5 to 50 psi. A formula is given correlating measured wear rates with friction and material properties. A mechanism of wear was evolved based on this correlation and on the experimental study of friction and wear surface-temperature fluctuations using special transducers and color motion picture photography. The predominant wear mechanism in the high-speed sliding of ceramic and cermet materials appears to involve formation of hot spots at asperity contacts and subsequent fracture of the material near these hot spots as a result of thermal stresses. In this situation, the wear rate is influenced both by the configuration of the rubbing parts and by the thermal-stress-resistance properties of the materials. When one of the mating surfaces is interrupted, the total wear is greater. Wear tends to predominate on parts with interrupted wear surfaces, such as slider bearings, in comparison with their continuous mating surfaces or ring surfaces, such as face seals, and on ceramic and cermet materials with low thermal-stress-resistance factor and low thermal diffusivity. (Source: U.S. Department of Commerce, Washington, D.C.)

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NEW PRODUCTS

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NEW JOURNAL BEARING DEVELOPED

A new Permwick Bronze Journal Bearing developed by Tann Bearing Co. division of the Tann Corp., Detroit, Mich., is introduced as having lifetime lubrication and a high life expectancy, and is interchangeable with general purpose ball bearings in rotating shaft applications.

The sintered bronze bearing assembly is designed for light to moderate loads at moderate speeds. It has a derated PV value of 25,000.

The Permawick Bronze Bearing is capable of operating in the temperature range of −25 F to 200 F. It is not recommended for stationary shaft applications, since centrifugal force may force oil through the sealed outer face.

The patented design uses a one-piece sintered bronze combination bearing and housing. Space between the members is packed with Permawick self-wicking lubricant which, in conjunction with an arrangement of oil slingers and closures, achieves a lifetime recirculatory oiling system.

Circle No. 1 on Reader Service Card

HOT GAS RELIEF VALVES

Two newly developed valves, designed for pressure control or safety protection in hot gas systems of liquid or solid propellants has been announced by Circle Seal Development Corp., Anaheim, Calif.

Both valves are constructed to control pressures generated by hot propellant gases. Specific uses are in auxiliary power systems, attitude control systems and test cells. A typical use results from problems associated with variations in solid propellant burning rates due to ambient pressure and temperature changes. Excess pressure resulting from higher burning rates is automatically vented by the Circle Seal valves to maintain predetermined pressure levels.

The Circle Seal V-4000 uses a gas reference pressure for relief pressure control. This unique design feature permits ready adjustment of relief setting from a remote source—a particularly useful feature in test cell applications. Relief pressure capabilities range from 10 to 1500 psi with temperatures to 2200 F. Suitable for air, nitrogen, combustion or decomposition gases.

The Circle Seal V-4001 is recommended for operational rocket-missile applications. The valve features an external spring which permits rapid dissipation of heat to eliminate cracking pressure variation which would otherwise result from excessive discharge temperatures. The virtually flat flow characteristics enable the valve to be used for pressure regulation as well as pressure relief. In operation the valve opens and closes at high cycling rates to insure that carbon particles which would normally collect in the seat will be broken off and blown clear. Pressure and temperature maximums and service applications are the same as the V-4000.

Circle No. 2 on Reader Service Card

NEW FLOW METER AND FILTER DEVELOPED

The new Flow Meter and Filter, developed by Capital Engineering and Manufacturing Co., Chicago, Ill., can be set to turn off power to equipment until filter is cleaned or replaced. Capital's Guardian Filter saves time and protects costly machinery. Being entirely automatic, a warning light appears when the filter becomes clogged and the liquid flow falls below a pre-set volume.

Circle No. 3 on Reader Service Card

A NEW BALL BEARING SCREW ANNOUNCED

A new line of ball bearing screws has been introduced by the Actuator Operation, Saginaw Steering Gear Division, General Motors Corporation, Saginaw, Michigan, to permit limited travel of the nut at predetermined positions with continuing rotation of the screw. This freewheeling action is accomplished by providing a stop in the screw at each end of the desired nut travel, and a simple freewheeling device as a part of the nut assembly.

Circle No. 4 on Reader Service Card

NEW ELEMENT DESIGN FEATURED IN MICROPOROUS FILTERS

A new conical pleated filter element developed by Microporous Filter Division, Anaheim, California, combines the advantages of reduced pressure drop and improved distribution of flow with minimum package size.

As flow passes within the filter housing, it is diverted by the streamlined nose piece and flows smoothly through the conical element. As flow passes through the element, flow area outside the element decreases as the flow inside the element increases, thus equalizing the flow pattern through the element.

The pleated element configuration provides an optimum contaminant holding area with minimum pressure drop and minimum package size. Conical element reinforcing is provided to withstand full system pressure. The unitized plug assembly is spun over to effect a perfect seal without the additional weight of a separate threaded plug and O-ring assembly.

Circle No. 5 on Reader Service Card

NEW WATER SOLUBLE MOLY LUBRICANT AVAILABLE

A new water soluble Molybdenum disulfide lubricant, used as an anti-wear additive for metal working, as a release agent, and as a lubricant pretreatment, has been developed by Bemol, Inc., Boston, Mass.

Known as Bemol #110 Suspension, the newly developed compound can be added directly to soluble fluids or can be diluted with water to form a drying lubricant pretreatment coating.

The #110 Suspension is said to be non-corrosive to all metals, plastics, and rubber. It can be used without decomposition to 750 F, and to 1,600 F for short periods in such operations as extrusion, forging, and molding. The manufacturer states that tests have shown that the product resists wear at pressures up to and beyond 400,000 psi.

Circle No. 6 on Reader Service Card

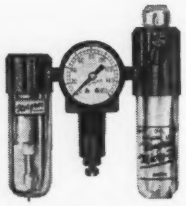
(*Continued on page 265*)

Norgren Lubro-Control Units

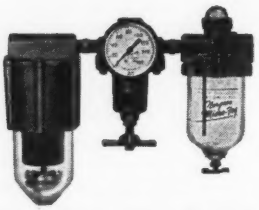
*A complete line of **visible oil feed** lubricator units for air-tools, air cylinders and other air-powered equipment*

NORGREN Micro-Fog Combination Units

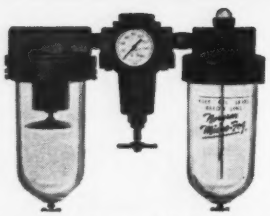
Easily adjustable for the exact amount of lubrication required



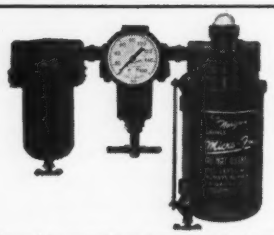
Transparent bowl filter; pressure regulator with gauge; transparent bowl Micro-Fog Lubricator. ⅛" pipe size. ½ oz. lubricator capacity. Small, compact unit for use where space is tight.



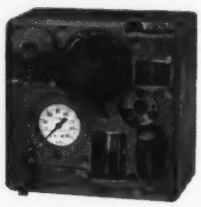
Transparent bowl automatic-drain filter; regulator with gauge; transparent bowl, Micro-Fog Lubricator. ¼", ⅜", ½", ¾", 1" pipe sizes. ½ pt. lubricator capacity.



Transparent bowl manual-drain filter; pressure regulator with gauge; transparent bowl, Micro-Fog Lubricator. ¼", ⅜", ½", ¾", 1", pipe sizes. 2½ pts. lubricator capacity.



Replaceable metal bowl manual-drain filter; pressure regulator with gauge; replaceable metal bowl, Micro-Fog Lubricator with constant oil level. ¼", ⅜", ½", ¾", 1" pipe sizes. 1 qt. lubricator capacity.



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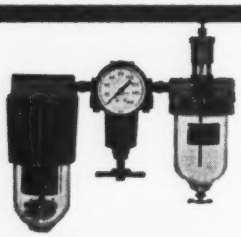
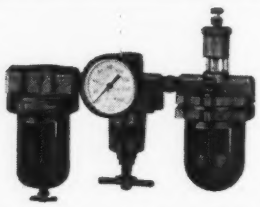
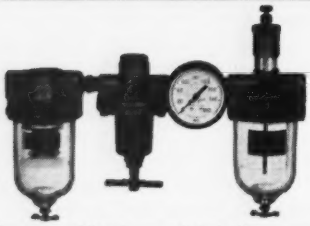


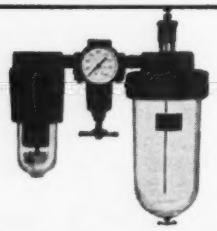
Transparent bowl automatic-drain filter; regulator with gauge; transparent bowl lubricator. ¼", ⅜", ½", ¾", 1" pipe sizes. ½ pt. lubricator capacity.



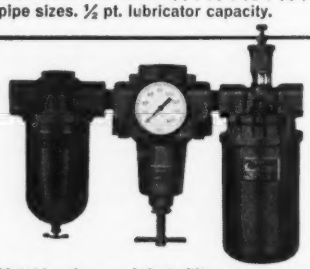
Replaceable metal bowl manual-drain filter; pressure regulator with gauge; replaceable metal bowl lubricator. ¼", ⅜", ½", ¾", 1" pipe sizes. ½ pt. lubricator capacity.



Transparent bowl manual-drain filter; pressure regulator with gauge on tee; transparent bowl lubricator. ¼", ⅜", ½", ¾", 1" pipe sizes. ½ pt. lubricator capacity.



Transparent bowl automatic-drain filter; pressure regulator with gauge; transparent bowl lubricator. ¼", ⅜", ½", ¾", 1" pipe sizes. 1½ qt. lubricator capacity.



Metal bowl manual-drain filter; pressure regulator with gauge; replaceable, metal bowl lubricator. ¾", 1" pipe sizes. 1 qt. lubricator capacity.

Shown here are a few models from Norgren's complete line of air filters, pressure regulators and lubricator units. For full information, call your nearby Norgren representative listed in your phone directory, or write for descriptive literature.

C. A. NORGREN CO.

3434 SOUTH ELATI STREET • ENGLEWOOD, COLORADO

NEW PRODUCTS

(Continued from page 263)

PILOT CONTROLLED AIR PRESSURE REGULATORS AVAILABLE

Remote regulation of inaccessible air lines is made possible by the new Pilot Controlled Regulator, manufactured by the Hannifin Company, Des Plaines, Illinois, a division of Parker-Hannifin Corporation.

In the Pilot Controlled Regulator, a column of controlled pressure air replaces the adjusting screw and control spring used in standard regulators. This air column applies a constant force on the diaphragm, and is governed by a pilot regulator operating at practically now-flow to assure prompt response, highest air flow rate and accurate pressure control.

Circle No. 7 on Reader Service Card

OIL FILTER-COMBINATION RECLAIMER MANUFACTURED

Unit consists of a motor pump set of practically any capacity required for forcing the oil through the HILCO Filter with either all of this oil or part of it being diverted to the HILCO Oil Reclaimer for final polishing and removal of moisture, water and other volatile contamination.

Flexibility of operation permits the unit to be used for restoring practically all types of lubricating and industrial oils to new oil condition. Various types of filtering materials are available to handle specific types of oil. Applications include internal combustion engines, compressors, vacuum pump lubricating and sealing oil, turbines and all types of electrical insulating oils.

Circle No. 8 on Reader Service Card

BONDED GRAPHITE COATING INTRODUCED

Superior Graphite Co., Chicago, Ill., has introduced a new bonded graphite coating. Called Slip-plate, this material comes in liquid form and can be brushed, sprayed, or rolled on. The product air-dries quickly to provide rust resistant protection as well as lubrication. It is said to be unaffected by temperature changes.

Circle No. 9 on Reader Service Card

AUTOMATIC FILL VALVE FOR FORCE FEED LUBRICATOR ANNOUNCED

The Lubricator Division of McCord Corporation, Detroit, Michigan announces the availability of the new No. A 30700 Automatic Fill Valve for Force Feed Lubricators.

The new valve is said to keep all oil reservoirs filled automatically. It is used for automation of gas engines and compressors, eliminating the need of manually filling the lubricator reservoir. The valve design is readily fitted to existing lubricators by top mounting with two screws.

This automatic fill valve is beneficial by preventing the lubricator from running dry and extends time period for checking oil supply on the Force Feed Lubricator. It is made of steel with ball check valve that lifts from its seat by the action of unicellular foam rubber float.

Circle No. 10 on Reader Service Card

GEAR PUMP GIVES HIGH CORROSION RESISTANCE

A new gear pump, designed in a thermoplastic material, offers high corrosion resistance in oil field applications. The product is cited as having good potential use in salt water disposal systems and in pumping wells where high salt water to oil ratios are found. Manufactured by Planet Products of Chicago, the pump uses Du Pont's "Delrin" acetal resin for all major parts.

Additional key operating advantages include very low heat build-up and almost noiseless operation.

Various models of the pump have capacities from 69 to 635 barrels per day and each is designed to operate at pressures of from zero to 100 psi with non-lubricating liquids.

Patented design features of the new Planet gear pumps are based on removal of the driving thrust load from the shaft and distribution of operating stresses over unusually large bearing areas—areas four times greater than normal.

A ball bearing unit inserted in the drive sheave is supported by the housing through the external hob on the front plate. The gears of "Delrin" have integral hubs over which a thin metal sleeve is pressed. The drive gear's hub runs in the bushings of "Delrin"; the idler gear's hub rotates in the actual housing cavity. This particular design eliminates need for an idler gear shaft, and does away with recessed bearings in the front or rear end plates.

The bearing characteristics of "Delrin" are demonstrated in the internal bearing gear supports. A thin-wall metal sleeve of brass, bronze, stainless steel, or other metal is pressed over the hub. It then rotates against a surface of "Delrin." The resulting combination of metal-to-resin, instead of metal-to-metal, requires no lubrication, eliminating need for grease fittings.

Circle No. 11 on Reader Service Card

NEW HYDRAULIC CHECK VALVE ANNOUNCED

A new Auto-Ponent hydraulic or air system check valve, incorporating a super-sensitive ball check in a forged body to provide full flow in one direction and instantaneous checking in the reverse direction, has been developed by Auto-Ponents, Inc., Chicago. Since the area of the ball is double the seat area, pressure applied over the greater ball area as it leaves its seat causes it to accelerate, giving a rapid opening at low differentials, without chatter.

The new check valves are said to have oversize internal passages and unrestricted ports to insure maximum oil or air flow and minimum pressure drop through the valve.

Circle No. 12 on Reader Service Card

NRC ANNOUNCES NEW 4-INCH VACUUM DIFFUSION PUMP

A new 4-inch fractionating oil diffusion pump, developed by NRC Equipment Corporation, Newton, Mass., is said to provide 120% higher speed over a one-third wider vacuum range than earlier models, offering lower backstreaming, higher forepressure tolerance, greater capacity, reliable operation, and minimal maintenance.

Designated Model HS 4-750, the pump has improved jet assembly and boiler designs which assure operation under widely varying conditions, the manufacturer stated. The pump speed peaks at 750 liters per second (1600 cubic feet per minute) between pressures of 1×10^{-3} mm Hg and 3×10^{-5} mm Hg and its total operation range extends from 450 microns to less than 1×10^{-9} mm Hg. Fore-pressure tolerance is 650 microns at blank-off. Backstreaming is less than 0.02 mm/cm²/min.

Circle No. 13 on Reader Service Card

(Continued on page 269)

Purchasing Agent, ". . . purchasing fewer types of lubricants in greater volume lowers our costs."

Superintendent, ". . . simplifying anything, including lubrication, pays off in improved efficiency."

GULF MAKES THINGS RUN BETTER

These four men understand the real value of a simplified lubrication program. *Why?* Because a minimum number of lubricants makes their jobs a lot easier.

In some plants, Gulf engineers have found more than *three* times the number of lubes actually needed for efficient operation. This means three times as many ordering, billing, stocking and handling problems. And it also means three times as many chances for human errors.

Examples: One large electrical appliance manufacturer slashed lubricants from dozens to just 6. In addition, a well-known paper company pared lube brands from 39 to 13.

Here's what Gulf engineers can do for you: (1) they can relate their experiences with hundreds of lubricating problems to *your* specific needs, and (2) they can recommend these versatile Gulf® lubricants—Gulf Harmony® Oil, Gulfcrown® Grease

Plant Engineer, ". . . a simplified lube inventory means reduced paperwork, fewer handling and stocking problems."

Oiler, ". . . fewer lubes mean less chance for error, and I get the job done faster."

THROUGH SIMPLIFIED LUBRICATION

and Gulf E.P. Lubricants.

Perhaps there are opportunities in *your* plant for simplified lubrication. A Gulf Sales Engineer is available to help you find them. Contact him at your nearest Gulf office. Or, write today for pamphlets on Gulf multipurpose lubes.

GULF

GULF OIL CORPORATION
Dept. DM, Gulf Building
Houston 2, Texas

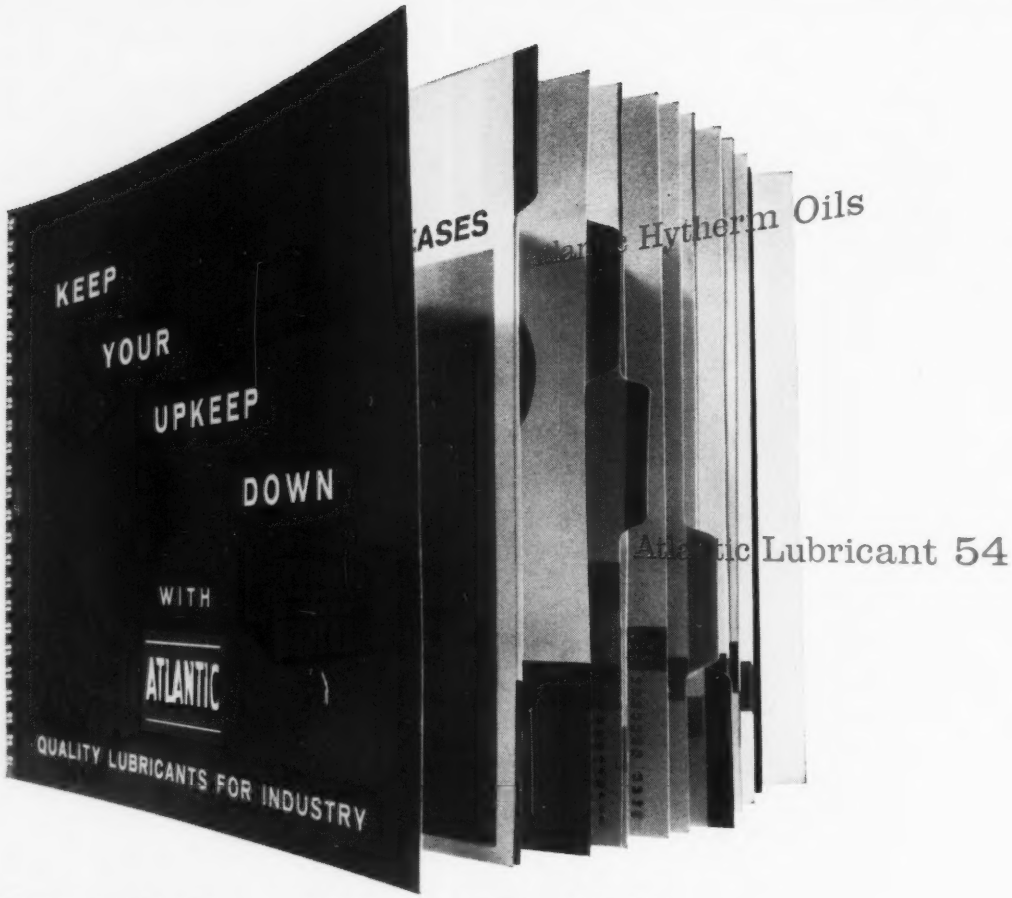
Send me booklets on Gulf multipurpose lubricants.

Name_____

Company_____

Title_____

City_____ Zone____ State_____



TWO PAGES...

in Atlantic's new booklet can solve 80% of your lubrication problems

These pages outline the properties and applications of Atlantic Hytherm Oils and Atlantic Lubricant 54 . . . two products designed to fill 80% of your plant lubricating needs. For your more specialized requirements, you can choose from a long list of quality Atlantic oils and greases. Result: *simpler purchases, use and storage.*

ATLANTIC HYTHERM OILS reduce wear, have good water separating properties, high flash point, low pour point and high Viscosity Index. ATLANTIC LUBRICANT 54 offers excellent water repellency combined with outstanding anti-rust properties. It is particularly effective where moisture conditions require extra protection against rusting and lubricant washout. An experienced Atlantic Sales Engineer will be happy to show you how these versatile, hard-working lubricants can fit into your plant operations.

The valuable 60-page booklet shown above is *free* to lubrication engineers and purchasing agents. It puts complete information on all Atlantic greases and oils at your finger tips. For your copy, simply write The Atlantic Refining Company, 260 South Broad Street, Philadelphia 1, Pa., giving your title and company name. We'll see that a booklet reaches you without delay.

ATLANTIC

THE ATLANTIC REFINING COMPANY

Philadelphia, Pa. Providence, R.I. Syracuse, N.Y.
Charlotte, N.C. Pittsburgh, Pa.

NEW PRODUCTS

(*Continued from page 265*)

PVC BALL VALVE ANNOUNCED

Walworth Company, New York, has announced the extension of its present line of PVC valves and fittings to include a type I (Normal Impact) polyvinyl chloride (non-toxic, non-flammable) ball valve. This corrosion-resistant, low-torque valve requires only a quarter-turn to operate.

Information is extensively discussed in a new 4-page bulletin now available. The valve is shown in a large cutaway view which pinpoints the outstanding features. It is manufactured in sizes ¼ through 4-inches and a table in the bulletin gives all the pertinent dimensions. Available in socket weld or threaded ends, the valve, which is virtually non-aging, features a rugged, compact body with external one-end adjustment.

Circle No. 14 on Reader Service Card

NEW, LIGHT-WEIGHT, PORTABLE COOLANT PUMP ANNOUNCED

"Mr. Cool," a portable coolant pump, manufactured by M-M-A, Inc., Lancaster, Pa., is usable on one or two machines at a time, with no plumbing or special mounting requirements necessary. Two outlets are provided for use on either one or two machines. Ample capacity is assured by use of a GE-¼ hp motor and many patented construction features which insure long life and efficient operation.

Complete details on construction, applications, capacities, prices, are available.

Circle No. 15 on Reader Service Card

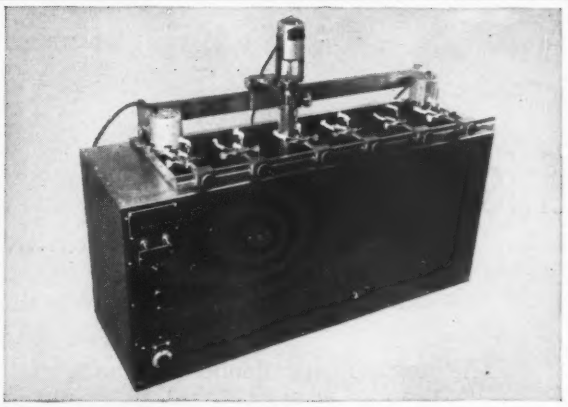
DRY FILM LUBRICANT AND RELEASE AGENT DEVELOPED

DRI-LUBE, a new dry film lubricant and release agent, is now available from Fluoro-Plastics, Inc., Philadelphia, Pa.

The product, packaged in an aerosol container, contains a fluorocarbon dispersion which puts an end to sticking, freezing, galling, slip-stick and abrasion. It is completely inert to nearly all acids, alkalies and solvents, and, is resistant to oil and water.

The slick surface it provides will remain through temperatures of −120 F to +500 F. DRI-LUBE is greaseless, doesn't collect dirt or dust and is nonflammable.

Circle No. 16 on Reader Service Card



New RAC unit yields demulsibility characteristics of lubricating oils

Utilizing the Wheeling Steel Test Method, this compact new RAC Demulsibility Bath accurately determines the three important factors in establishing lubricating oil characteristics: percentage of water in the oil, total ml. of "free water," and ml. of emulsion separated from the mixture.

The all stainless steel bath is well insulated to prevent heat losses, offers two continuous duty stirrers for even heat distribution and Hevi-duty electrical heaters that provide temperature range from 50° to 250°F. Available in six, four and two-sample units, bath has inside dimensions of 36" X 12" X 18" and is equipped with a variable speed emulsifier with maximum of 8000 RPM. Unit can be easily converted to a regular water bath for other test work. Write for FREE bulletin giving specifications and prices.

RESEARCH APPLIANCE COMPANY

Box 307 • Allison Park, Pa.

NEW AND EXCLUSIVE with McCord Model 55 Lubricator

SAFE-LEVEL SIGHT FEED

That's right! The finest lubricator on the market today has been refined to give you even *more* for your money. Now, the McCord Model 55 Lubricator comes equipped with new, exclusive Safe-Level Sight Feed with these outstanding benefits for you . . . safe, sure warning of malfunction . . . quicker priming . . . better feed regulation . . . extreme accuracy . . . improved reliability. IMPORTANT: Safe-Level Sight Feed can also be added to existing Model 55 Lubricators.

For COMPLETE details on this important new development write today to:

McCORD CORPORATION

Lubricator Division • Detroit 11, Michigan

■ *Third in a series of articles designed to promote broader understanding of colloidal dispersions.*

try this on for size

The increasing use of solid lubricants, particularly as dispersions in oils and greases, has focused greater attention on the "size" of the solid lubricant particles. For particle size is one of several important characteristics of a dispersion which may suggest its use for specific applications in a manner which is both technically and economically sound. It is good that a user or potential user of a lubricant dispersion should show this interest. Unless this subject is considered carefully, serious misconceptions and misapplications may result.

Fundamentally, the term "particle size" has meaning only as it refers to a particular particle of a particular shape. As applied to true spheres, it is quite specific; when used to describe the flake-like particles of colloidal graphite or molybdenum disulfide the expression requires some clarification in order to be meaningful. Claims and specifications should be clear as to the exact physical dimension being described. Is it the longest distance across the face of the flake? Is it the thickness, or is it the diameter of a sphere of equivalent volume? It is important to distinguish between these values as they may be significantly different.

Dispersions of the type which we produce consist of stabilized particles having a *range* of sizes. Therefore, it is necessary to describe the limits of this range and the characteristic distribution of the total particles. This is portrayed by means of a size distribution analysis and from this, an "average" size is frequently calculated. But this expression is useful in describing dispersion quality only when the factors of shape, distribution, and specific test method are included.

Methods used to determine particle size should be appropriate to the task. Techniques applicable to powdered materials may yield very faulty data if applied to dispersions of the same powders. Procedures well suited to certain solids can be highly misleading when employed for others.

At Acheson Colloids, fine particle technology is our business. Procedures, equipment and terminology are carefully selected and employed to accurately measure and describe the materials with which we work. This careful attention to detail has been a major factor in building the confidence expressed by users of Acheson dispersions throughout the world.

ACHESON—First name in solid lubricants for fifty-three years

ACHESON **Colloids Company,** PORT HURON, MICH.

A division of Acheson Industries, Inc. Also Acheson Industries (Europe) Ltd. and affiliates, London, England. Sales offices in principal cities.

PRODUCT LITERATURE

The Society will not be responsible for the accuracy of statements made in this column. Any reference to proprietary products does not imply an endorsement of such products by the Society.

NEW LOW-COST LIQUID BLENDING DESCRIBED IN BULLETIN

Vari-Flo proportioners and new concepts in continuous line blending are detailed in Bulletin 650, released by Blackmer Pump Company, Grand Rapids, Mich.

According to the bulletin, Blackmer employs the flow control principle of their Vari-Flo variable volume pumps to develop complete proportioning systems which offer substantial savings over alternate blending methods. The new equipment is designed for applications involving total flow rates between one and 2000 gallons per minute, with larger systems possible by adding more components.

Simplicity of design and operation and lower initial costs are said to be the chief advantages over other line blending equipment now available.

Also included is a series of brief factual reports from typical users documenting the accuracy and economies claimed for Vari-Flo line blending. These on-the-job case histories are complete with pictures and detailed diagrams of each installation. The bulletin has an outline of basic information required by Blackmer engineers to quote on a given application.

Circle No. 17 on Reader Service Card

TORQUE MEASURING INSTRUMENTS BROCHURE

A new 12-page brochure, entitled "Torque and Efficiency Measurements," illustrates how to measure the torque characteristics and speed of motors, gear trains, servo mechanisms, and potentiometers. Covers torque ranges from ¼ gm-cm to 200 lb-in.

The brochure, published by Power Instruments, Inc., Skokie, Ill., includes formulas for computing power and efficiency and methods of using stroboscopes and tachometers for analyzing rotating motion.

Upon defining torque terms, the brochure discusses torque testing methods and gives specific set-ups and recommendations.

Circle No. 18 on Reader Service Card

BULLETIN DESCRIBES PRESSURE REGULATORS

Specifications on Reliance Type HPC Pressure Regulators for reducing applications are described in new Bulletin 123, released by American Meter Company, Erie, Pa.

Designed with secondary guides and isolator plates, both types are recommended as primary regulators for use with ammonia and for general service in gasoline plants and refineries. Bulletin 123 describes three models; 101—spring loaded, 102—pilot loaded, and 102 DB—pilot loaded with downstream bleed. Cutaway illustrations of all three show the internal design features. Complete descriptions, weight, dimensions and capacities are given.

Circle No. 19 on Reader Service Card

SILICONE INFORMATION BOOKLET PUBLISHED

A new and comprehensive booklet on silicones containing advanced information on these important man-made chemicals has been issued by the Silicones Division of Union Carbide Corporation, New York, as an aid to engineers and technical personnel.

Graphically illustrated with photographs, charts and graphs, the booklet goes into detail about what silicones are, describes their manifold uses for consumer and industrial products, and suggests ways in which they can be adapted to a host of new applications by the design engineer or product development manager. Of special significance is the series of charts covering the properties and features of Union Carbide Silicone fluids, resins, rubber compounds, water repellents, antifoams and emulsions, and their adaptability for use by the aviation, automotive, chemical, electronic, rubber, paint, paper, textile, glass, metal working and other industries.

Circle No. 20 on Reader Service Card

RUST PREVENTIVE BROCHURE RELEASED

The new Brochure No. 611, published by Rust-Lick, Inc., Boston, Mass., covers features, applications, and packaging of Rust-Lick 606, a specially formulated spray-on, moisture-absorbing, lubricating mist which can be used on wet or dry surfaces and which is said to provide positive long-term protection against rust.

Circle No. 21 on Reader Service Card

DATA SHEET ILLUSTRATES APPLICATIONS OF SOLID-FILM DRY LUBRICATING STICK

Typical extreme pressure applications of MOLYKOTE Lubricating Sticks are the subject of a data sheet recently published by The Alpha-Molykote Corporation. According to the data sheet the lubricating stick, which contains molybdenum disulfide solid lubricant in a resin binder, is particularly useful on grinding wheels and cutting tools. Other applications described include tools for cutting other materials such as paper and wood, friction surfaces of all types and, when machined to shape, as an actual self-lubricating bearing element in small machines. In addition, the two-page, two-color bulletin also gives tips for effective application.

Circle No. 22 on Reader Service Card

BOOKLET ILLUSTRATING VALVES, FITTINGS, AND TUBING AVAILABLE

A 20-page booklet illustrating the complete line of Terriss stainless steel sanitary fittings, valves and tubing is now available from Terriss-Consolidated Industries, New York. Terriss-Consolidated Industries fabricate corrosion resistant equipment.

Circle No. 23 on Reader Service Card

Tennessee Coal & Iron · Weirton, Steubenville · Irvin Works
Youngstown Sheet & Tube, Indiana Harbor · Columbia–Geneva
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Steel Company of Wales, Trostre · Fuji Iron & Steel Company
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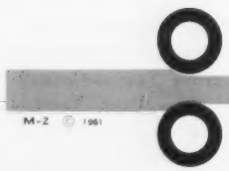
The Bearing Used Around the World for Tin Plate Temper Pass Mills

Forty-five two-stand tandem temper pass cold mills—90 stands total—are rolling superior tin plate on **MORGOIL** bearings the world over. Regardless of ultimate requirements for capacity, speed and stiffness **MORGOILS** are always your best choice.

The reason: **MORGOIL** bearings are products of many years' experience in the rolling mill field featuring sound design, precision manufacture by skilled craftsmen and continuing research—all backed by an alert field service organization.

Write for technical counsel on any application.



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MORGAN WORCESTER

MORGAN CONSTRUCTION CO.
WORCESTER, MASSACHUSETTS

ROLLING MILLS · MORGOIL BEARINGS · WIRE DRAWING MACHINES · COMBUSTION CONTROLS

The Northern California Section celebrated "Past Chairmen's Night" on March 16. Shown here is the speakers table with (left to right) Mr. Doug Godfrey, who introduced the speaker; Dr. Francis J. Clauss, the speaker; Messrs. Leonard Dreher, outgoing chairman for the 1960-61 term; Julian Le Sire, incoming chairman; George Castle, past secretary and incoming treasurer; and Howard Scheffel past treasurer and incoming vice chairman. In the foreground (left to right) are Messrs. Isbell and Harvey Trowbridge, past chairmen.

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Jacques Sejournet
Cefilac
Paris, France

*Reinstated

ASLE European Study Trip Cancelled

W. L. Roberts writes us "It is with sincere regret that I must inform you that the proposed ASLE trip to Europe has been cancelled." The number indicating interest in the trip was too small to justify its undertaking.

VADEN VANISOUL

New Epoch making Additive in the preparation of Water-Soluble Grinding Fluid.

By adding 1% in volume of VADEN VANISOUL to water, a splendid grinding fluid, excellent both in anti-rust and machining capacities, will be produced.

This fluid guarantees the most precise and cleanest finish to the metal work surfaces, while improving the grinding efficiency.

(For further Technical Literature please apply to us)

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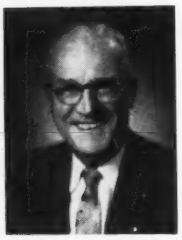

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Lubrication Engineering

the secret of success

You can't stand still today . . . it's either move ahead or fall behind. The marketing fraternity calls this "grow or die." Since its inception in 1944, ASLE has grown—in numbers, in services to members, in stature throughout industry. We've suffered growing pains too. ASLE has a few of these pains right now.

Some may term these pains "problems." I think they are more properly called "opportunities." Problems don't occur when an organization is standing still. They are a part of the process of progress. Their solution creates the opportunity to better serve the individual and industry.

No one of us can capitalize on these opportunities by himself. The true measure of ASLE will be how well we work together. Each of us can add his small part, and the total of these parts can be significant achievement.

This is how ASLE has grown, and it continues to be our secret of success. During this year, every member will be called on to contribute time and ability to the attainment of our goals. I know members will respond in the future as they have in the past—with enthusiasm.

D. M. Cleavelend
President, ASLE

Status Report—

Synthetic Lubricants

Editor's Note—

STATUS REPORT—SYNTHETIC LUBRICANTS is the second of a series of articles in various areas of general interest to lubrication engineers. Subjects have been selected on the basis of the reader preference survey conducted last fall.

In preparing these articles, your editorial board has been fortunate in securing the cooperation of nationally recognized authorities in various fields of lubrication research and engineering. Each article will comprise their comment in response to specific key questions on the subject under discussion.

J. Boyd

commentary by

William H. Millett
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William Eismann, Jr.
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In every branch of the sciences these days, glamorous phrases capture the imagination but have a tendency to leave the mind on a loose tether. With "synthetic lubricants", lubrication engineering is no exception.

Yet there is so much of real importance and value implied in the term "synthetic lubricants" that it must not be allowed to remain vague, perhaps to deter serious consideration of many advanced materials. Rather, the term must be given specific meaning and made a useful part of the lubrication engineer's working vocabulary.

Accordingly, here are the answers to a series of practical questions on the subject. Dr. William H. Millett and William Eismann, Jr. have both spent the greater part of their careers in the study and development of synthetic lubricating materials and are nationally recognized as authorities in the field.

Naturally, this discussion of a subject so broad can be neither complete nor definitive. Instead, it is intended to bring synthetic lubricants into sharper focus and facilitate the more detailed investigations that must precede actual application.

Question: What does the term "Synthetic Lubricant" mean?

Answer: Actually, the phrase itself is so broad as to be virtually devoid of real meaning for technical purposes. It covers many different types of materials, selectively suited to a wide range of applications and produced by a variety of processes. In lubrication

engineering usage, however, it has acquired a series of practical interpretations depending on the context in which the phrase is used.

First, and most generally, the phrase is used to distinguish between man-made lubricants and the naturally occurring or derived hydrocarbon materials. Many of the former are, of course, at least partially derived from petroleum but are fundamentally altered by chemical means. Typically, synthetic lubricants as a group are either definite chemical compounds or mixtures of compounds of the same chemical class. Petroleum based hydrocarbon lubricants, by contrast, are almost always a mixture of hydrocarbon fractions of different chemical structure. The addition of various additives of property improving substances does not make these hydrocarbon materials synthetic.

The chemical uniformity of synthetic lubricants permits sharper definition and more precise manipulation of their physical properties—and leads to a second generally applied meaning for the term "synthetic lubricants". It connotes materials produced to meet special operating requirements beyond the normal capabilities of hydrocarbon lubricants. High temperature stability, high viscosity index and fire resistance are among the most important of these special requirements.

As a matter of fact, these extended performance needs originally gave rise to the development of synthetic lubricants when, shortly before World War II, it became apparent that conventional lubricants would not permit the tremendously improved mechanical performance demanded in certain military equipment. Since then, military needs have continued to be the principal force behind synthetic lubricant development. But despite this military orientation, the synthetics have become very important in industrial lubrication and are gaining wider and wider application.

Third, and most specifically, the term "synthetic lubricants" is actually a family name for diverse groups of chemical compounds including silicones, polyglycols, phosphate esters, di-esters and polyesters, halogenated hydrocarbons and silicate esters. Mixtures of synthetic and petroleum based hydrocarbon materials are also sometimes considered as a sort of foster child in this family but really constitute a separate subject beyond the scope of this discussion.

To get down to cases, any discussion of synthetic lubricants in general must sooner or later be narrowed down to one or another of the above materials. They differ greatly and there can be no real evaluation of properties versus requirements until the general terms have been put aside in favor of the name of a specific chemical compound or type of compound.

Question: How do these various types of synthetic lubricants differ?

Answer: About as widely as any group of fluids possible could and still be considered a family—in cost, composition and application. In fact, fluidity and use for lubricating purposes are really about the only common traits. Rather than attempt to describe the differences, which could imply a primary degree of similarity, here is a run-down on the outstanding characteristics of the various types of chemical compounds ordinarily accepted as members of the synthetic lubricants family (the order of presentation here has no significance):

1. *Silicones* . . . are chiefly known for their great resistance to high temperature degradation and high viscosity index, in excess of 200 by conventional standards. As a lubricant, especially in certain steel on steel applications, they leave something to be desired.

2. *Polyglycols* . . . are the least costly of the synthetics and combine virtually all of the desirable characteristics of synthetic lubricants. But, like the jack-of-all-trades, the polyglycols are exceeded in many characteristics by more specialized materials.

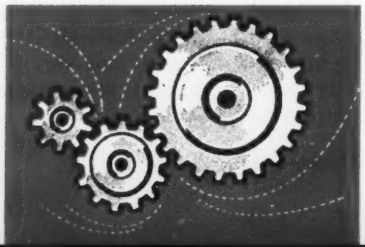
3. *Phosphate Esters* . . . are highly fire resistant and provide excellent lubrication but are limited in application by a relatively low viscosity index with resultant limitations in low temperature service.

4. *Di-esters and Polyesters* . . . remain liquid over an extremely wide temperature range and, as the principal lubricating material for jet aircraft, are the synthetics used in the greatest quantity. Di-esters are restricted, however, to a relatively low viscosity range.

5. *Halogenated Hydrocarbons* . . . offer extreme stability and resistance to both fire and high temperature breakdown. Viscosity index and low temperature limitations as well as system compatibility are the biggest drawbacks here.

6. *Silicate Esters* . . . combine good lubricating qualities with high temperature stability and are chiefly used in high temperature hydraulic systems operating over wide temperature ranges. The strong tendency of some of these compounds to interact with water or moisture is their weakness.

All of the above is highly generalized and only serves to emphasize the fact that the various materials comprising the family of synthetic lubricants do indeed differ widely and must be considered very carefully for each specific application. By no means should these thumbnail descriptions be allowed to inhibit thorough exploration of any material for any use. Actually, the application of all of these substances to various industrial needs is still in the creative art stage.



Question: When should synthetic lubricants be considered for use?

Answer: Without wishing to convey this impression that the synthetics are used only in desperation, the simplest and most direct answer to this question is, "When conventional petroleum based hydrocarbon lubricants fail." In considering synthetic lubricants for any industrial application, the fact that they are considerably more expensive than conventional materials must be faced and their use must be economically justified by some direct requirement for the advantages that they alone can offer.

In equipment design for instance, when an impasse between space limitations and bearing or gearbox sizes looms, using one or another of the synthetics will often permit a workable compromise. In general, because they will work at higher temperatures and for longer periods, the use of synthetics will often result in improved performance. This is usually ample economic justification for the use of the more expensive lubricant.

In everyday plant operation, heavier and heavier demands on existing equipment, leading to accelerated wear and frequent breakdowns, is a common pattern. Synthetic lubricants should certainly be considered in such cases because they can actually extend the capabilities of existing equipment. "Running hot" is invariably a warning of trouble with hydrocarbon lubricants. The use of the proper synthetic lubricant can often improve this condition.

Two other general situations usually indicate the use of synthetic lubricants. These are where the elimination of fire or explosion hazards is critical and where equipment must operate unattended for long periods of time or under extremes of temperature variation.

The mere availability of the various synthetics, with the superior fire resistance common to some of them, has helped to concentrate attention on the hazards created by the use of flammable hydrocarbons in lubricating compressor and hydraulic systems. Where these hazards were formerly accepted as unavoidable, they now constitute a calculated risk and insurability alone will often justify the use of fire resistant synthetic fluids.

The use of equipment designed for long-term service without recurring short-term maintenance is growing more commonplace. Synthetic lubricants offer a higher degree of durability that will permit operation, even under difficult environmental conditions, for periods up to several years without attention. The stop light timing mechanisms that now function so dependably, year in and year out over a temperature range of more than 100° F., are frequently serviced with synthetic lubricants.

But to put a broader light on the consideration of the use of synthetic lubricants, it can be said that they add a new string to the lubrication engineer's bow. Problems that had to be endured before can be solved now, and there are almost unlimited opportunities for improving equipment performance and life or decreasing equipment maintenance costs. Thus, synthetic lubricants should at least be considered wherever a problem exists or where improvement is desired. Economics will ultimately determine whether or not they should be used.

Question: Where are synthetic lubricants now being used?

Answer: By volume, principally in aircraft where operating temperatures and related requirements make their application mandatory. The actual volume of industrial use is somewhat less, not because of a lack of need but rather because the synthetic materials are relatively new to industry and are still being evaluated.

Where synthetic lubricants have been applied to industrial operations, the results have frequently been dramatic. Here are a few examples:

1. The wheel bearings of kiln cars carrying ceramic products through a tunnel kiln, operating at about 500° F, formerly required lubrication twice during each 36 hour trip through the kiln and broke down frequently despite this care. With a synthetic lubricant, weekly lubrication suffices and the cars run for months without bearing failure. Further, the driving power required to move the cars dropped by almost half.

2. In a large hot air (about 630° F) circulating fan, ball bearings failed on two test runs in 10 and 12 days respectively with 2 different high quality hydrocarbon lubricants. The third run, using a synthetic lubricant, continued for over 15 months with no sign of trouble.

3. Quarterly teardowns involving almost complete replacement of bearings and gears, with frequent intervening breakdowns, were the rule for heavily loaded worm gear drives in a large textile plant. Simple substitution of the correct synthetic for the E. P. hydrocarbon lubricant previously used reduced the teardown schedule from 4 times a year to 1 and virtually eliminated breakdowns. Moreover, little or no replacement of parts was required at the time of teardown.

4. The characteristically destructive start-stop operation of lift trucks in a large plant made it necessary to overhaul their engines every 6 months. As a test, a synthetic crank case lubricant was installed in one engine, replacing a premium grade petroleum based oil, in lieu of a regular overhaul. Within a week of operation, the solvent action of the synthetic had cleaned up the engine, making an overhaul unnecessary, and it then ran without overhaul for 30 months until mechanical failure not traceable to lubrication failure.

In addition to such instances of dramatic cost justification, synthetic materials are probably finding their widest industrial application in hydraulic systems. The advantage here is fire resistance, but it is quite difficult to cite case histories on fires that didn't

occur. However, the simple fact that this type of application is growing by leaps and bounds, plus official cognizance of the fire resistant fluids by leading fire underwriters, is ample proof of the real economics involved.

Question: Are synthetic lubricants commercially available?

Answer: Yes, to meet virtually every requirement, and through the same channels of procurement as petroleum based hydrocarbon lubricants. Certain synthetic lubricants, though marketed in smaller volume, are available in bulk quantities. Others, principally because of the economics involved, are currently marketed by the pound or gallon rather than by the drum or carload.

The cost of synthetic lubricants is a significant factor, and the use of smaller quantities is almost a condition of application. Superior stability characteristics and the resistance of synthetics to breakdown or degradation in use, however, tends to make a little go a long way. Also, waste and loss almost invariably decrease, simply due to recognition of the value of the material by those responsible for its use.

While first cost is relatively high with the synthetics, the overall cost of use is often surprisingly low.

Along this line, synthetic greases are also available. Conventional soaps and thickeners are used in some of these and will give satisfactory performance up to about 300° F. Beyond this point, synthetic thickeners as well as lubricants are used and the range of operation is extended to 500° F or even higher.

The only real availability problem in connection with synthetic lubricants today is not *where* to get them but *what* to get. Until general experience and knowledge are far beyond their present state, the safest procedure in deciding what material to buy is to seek the counsel of specialists.

Question: Does the use of synthetic lubricants involve any special precautions?

Answer: Not in the sense that there are dangers or hazards involved, but it must be remembered that the synthetics are not just an extension or refinement of familiar petroleum lubricant technology. Rather, they comprise completely new and different classes of materials with completely different characteristics.

In this sense, certain precautions should be observed—just as in the case of any change of system or procedure. New habits must replace old ones when changing from the one type of lubricant to another in systems designed for use with petroleum oils.

The most important aspects of synthetic lubricants that involve precautions in this respect are the increased possibility of incompatibility with other operating fluids or lubricants and incompatibility with system components such as seals, packings, paints, filters and hoses. Once the decision to use a given synthetic lubricant in any specific application has been made, all of these factors must be considered.

The cost of the material lost alone makes leakage a serious matter. Therefore, where some degree of leakage is often tolerable and even considered normal in systems using petroleum based lubricants, joints must be tight when synthetic lubricants are used. Even more important, though, is the fact that certain synthetics are hard to hold. System overhaul is generally a good idea before changing to a new synthetic lubricant. This also provides an opportunity to replace old seals, hoses and packings with components made of specially resistant materials. Various of the synthetics have strong solvent characteristics and will attack many of the materials that successfully resist decomposition in the presence of hydrocarbon lubricants or fluids.

This powerful solvent action will also affect many of the paints and finishes in common use in industry and suitable action to correct this condition should be taken.

Incompatibility with other fluids is also common to many of the synthetic lubricants and makes very thorough cleanout of the system into which a synthetic is to be charged essential. Even trace amounts of foreign lubricating materials or water can have serious consequences if allowed to contaminate the synthetic material.

Beyond these basic and logical mechanical precautions, the lubrication engineer must also take into account the fact that many of the synthetics are quite new. The background of established technical data concerning performance in given types of applications is not always available.

Take hydraulic system applications for instance. Under certain circumstances, the higher densities of the synthetics may lead to excessive cavitation, and differences in compressibility may have an effect on performance. Relatively little data are generally available in some of these areas.

For all of this, though, the decision to apply synthetics involves no more special precaution than any other engineering change in a going operation. Actually, because the synthetic materials themselves are so precisely controlled and stable, many of the precautions or risks normally involved in a process change can be avoided.

Question: What effect will synthetic lubricants have on the technical aspects of lubrication engineering?

Answer: Mostly in the form of welcome opportunity. There are those, of course, who shudder when the old, familiar status quo is threatened, but the availability and increasing acceptance of synthetic lubricants will do a great deal to advance the science of lubrication engineering.

With synthetics, the practicing lubrication engineer has much greater latitude in approaching ordinary problems, a whole new bag of tricks to help him keep the plant running smoothly. Even more, he now has the means at his disposal to establish new levels of operating efficiency, actually to increase plant capacity.

But it won't be as easy as falling off a log. There's a lot of difference between new synthetic lubricants and the good old fashioned snake oil that cured all. To take advantage of what synthetics have to offer, an almost entirely new area of technology must be understood.

First, there are the capabilities and limitations of all of the different types of synthetics to consider. These differ widely and are far more complex than the well-known hydrocarbon pattern.

Then, there is a new additive technology to learn. Additives for the synthetics are often completely different from those in current use.

Some testing procedures will also require change. The standards which have served for many years to gauge the quality and predict the performance of hydrocarbon lubricants are not always applicable to the synthetics.

Summing up, synthetic lubricants will have a profound effect on the technical aspects of lubrication engineering, but a beneficial one that will certainly result in increased recognition and stature for the lubrication engineer. True, it will take some doing to master the proper application of synthetic lubricants but the game is worth the chase.

Question: Are there significant practical considerations from the lubrication engineer's viewpoint?

Answer: As distinguished from certain technical aspects of the synthetic lubricants as just discussed, the practical considerations would cover handling and storage, routine lubrication scheduling, inventory control and personnel training.

Taking these in order, storage and handling requirements are not essentially different from those of good maintenance practice. The synthetics as a class are very stable with excellent shelf life. Scheduling will be simplified merely because less frequent application and change of lubricating materials will be necessary.

On the other hand, both inventory control and the training of lubrication maintenance personnel will require a little more attention. There will be a tendency, at least for the near future and until the basic properties of the various types of synthetic materials are completely familiar, to apply synthetics by brand name rather than by specification—and the number of different materials in use in most plants will certainly increase.

Further, because the synthetics are not usually interchangeable and are quite sensitive to contamination, greater care in routine lubrication maintenance will be required. Workers performing these tasks will have to be more carefully trained and supervised than is usually the case. With 3,000 gallons of a five dollar per gallon material in a system, gambling on the loss of the entire quantity because of an error in adding a few gallons of the wrong make-up fluid will be too costly.

From the practical viewpoint, the lubrication engineer's job will change as synthetics are used more widely but should not be significantly more difficult. Simplification in some areas will offset the extra effort required in others.

Question: What is the current status of synthetic lubricants?

Answer: Synthetic lubricants, like opportunity, are knocking at industry's door, waiting for the lubrication engineers to open up. This should not be taken to imply that synthetic lubricants will replace hydrocarbons to any great extent, but they do offer solutions to a great many problems.

Actually, research in the synthetic lubricant field is far ahead of industrial application (less than 1% of all lubricants used by dollar volume) at this time, thanks to the pressure of military requirements. But usage is certain to grow. And as it does, because of the increased technical proficiency that application of the synthetics will make essential and the new benefits that they will bring, lubrication engineering will also grow in both stature and recognition.

To put it briefly, "ready, willing and able" describes the status of synthetic lubricants perfectly as far as industrial application is concerned. They await only acceptance—which will sooner or later be forced by the need for the properties that only the synthetics possess.

Studies on the Dispersion of **SILICONE**

Defoamer in Non-Aqueous **FLUIDS**

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The storage stability of dispersions of 5 to 30 ppm silicone in synthetic and mineral lubricating oils was studied as a function of the mixing method used in their preparation. Both radioactive tracer (H^3) and emission spectrography were used as analytical tools in following the separation of the additive; detailed procedures for these methods are given. It was concluded that the most stable dispersions were made by the use of high intensity mixing approaching colloid milling.

INTRODUCTION

Silicone fluids have been used as defoamers in lubricating oils and similar fluids since the late days of World War II (1). The literature contains several articles on the general behavior and physical chemistry of this system. However, there is one area of great practical importance in which little work appears to have been done. That is with respect to factors affecting stability of these dispersions during storage. Retention of foam inhibition properties is vital to satisfactory end use of the product. On several occasions, involving different types of products, there has been noticed a tendency to lose the antifoam properties upon storage. In the extreme case, this can result in a complete failure of equipment in certain critical applications. Unfortunately, it has been impossible in the past to obtain quantitative data on this phenomenon because of lack of suitable analytical methods for determining silicone fluids in the normal concentration range of two to thirty parts per million which is used. The results of foam tests are purely empirical, as they provide no quantitative measure of silicone concentration. Therefore, foaming data contribute little to a clear understanding of the general chemistry of the system. The present work has therefore been directed towards (a) developing suitable analytical and tracer methods to study this problem, and (b) their application to a study of the effect of mixing methods on the storage stability of silicone dispersions.

Revised version of a paper presented before the Division of Petroleum Chemistry, American Chemical Society, Cleveland Meeting, April 11-14, 1960.

EXPERIMENTAL PROCEDURES

A radioactive technique and an emission spectrographic method were developed for quantitative determination of silicone content of blends.

Radioactive Technique for Silicone Determination

The principal technique worked out for this study was the radioactive tracing of a typical silicone fluid. In order to do this manipulation as simply as possible, the Wilzbach technique (2) was selected. This method of tracing, which consists of exposing some of the material to a relatively large amount of pure tritium gas, has the characteristic of providing random labelling throughout the hydrogen atoms of the molecule. The alternative, which was the synthesis of the radioactive silicone, has the ability to trace atoms in a certain position. For the present purpose, either method would have been satisfactory; however, the synthetic method is much more expensive. The disadvantage of the Wilzbach method is that the energetic beta particle bombardment to which the molecules to be traced are subjected has a powerful degrading effect and, therefore, the Wilzbach reaction product inevitably requires substantial purification. The degradation products range from gaseous material to polymers. In the case of most additives, this can be a very serious matter. However, since the silicones involved are a homogenous series of uniform chemical structure and differ only in degree of polymerization, a certain amount of by-product can be tolerated relatively easily.

A one-gram sample of 350 centipoise dimethyl silicone (commercial grade) was exposed to ten curies of tritium on a shaker for two weeks. After the irradiated silicone was separated from excess tritium, it was purified by two simple steps. First, a sample was dissolved in toluene and filtered to remove high molecular weight polymer, of which there was an appreciable amount. The filtrate was then divided by atmospheric

and vacuum distillation into four cuts, the noncondensable gas being determined by difference. Results of this distillation were as follows:

TABLE 1.—Volatility Vs. Activity of Irradiated Dimethyl Silicone

Temperature	Pressure	% of original activity
100-150 C	1 atmosphere	25.6
150-240 C	2 millimeters	8.8
240-300 C	2 millimeters	21.6
Residue	2 millimeters	24.7
Loss	—	19.3

As all the original silicone boiled above 300 C at 2 millimeters, this indicates 75.3% of degradation products exclusive of insoluble polymer. The distillation was repeated several times using different viscosity grades of silicone as carriers. It was thus possible to obtain silicones, traced to a fairly high specific activity, of 10 centipoises, 350 centipoises and 60,000 centipoises at 25 C. The latter two were used in the mixing studies.

It was found that traced silicone could be determined with a considerable degree of accuracy at the concentrations involved by the use of a liquid scintillation counter. With the conventional counting fluid, the PPO-POPOP "Cocktail"*, an efficiency of about 20% was achieved. The principal complication from this part of the work was the tendency of the solution in the counting vials to change efficiency within the counting period. This was eventually traced to the formation of a very thin silicone adsorption layer on the glass vials which led to such extraordinary results as an apparent half-life of 44 hours for tritium, which is well known to have a half-life of 12.3 years. The difficulty encountered was completely eliminated by precoating all vials and other glassware used with a thin film of inactive silicone deposited from a 1% solution in toluene. The accuracy obtained on this method, on a single determination, was approximately ±.2 part per million of silicone on blends nominally containing 10 parts per million.

Emission Spectrograph Procedure for Silicone Determination

An emission spectrograph procedure which has been used with fairly good results for high concentrations of silicone was developed further to obtain the maximum degree of sensitivity available. This method is a general procedure for silicon when present as an organic or inorganic compound. Therefore, the procedure is not applicable to used lubricants containing extraneous silicon compounds which are present as silicates, etc. The method may be carried out on any direct reading spectrometer. The rotating disc electrode technique is employed (4); that is, the oil sample is held in an aluminum boat and introduced directly into the electrode gap as a thin layer of oil on the periphery of a graphite disc which serves as the sample electrode. Calibration standards were prepared by adding known amounts of soluble organic silicon compounds to the silicone-free base oil. Octaphenylcyclotetrasiloxane, available from the National Bureau of Standards, was found to be satisfactory. To avoid matrix effects, it is important that the base stock used be approximately the same composition as the oil to be analyzed. Concentrations of 0, 3.8, and 7.6 ppm Si were used.

High voltage excitation having the following parameters was used:

Primary Voltage	200
Discharges per Cycle	4
Capacitance, μf	0.007
Inductance, μh	410

The exposure conditions were:

Spectral Line	Si 2881.6Å
Internal Standard	Reflected Beam
Slit Width	
Primary	20 μ
Secondary	100 μ
Prespark	10 seconds
Exposure	35 seconds
Rotrode Speed	10 RPM

Working curves were prepared by exciting each standard in quadruplicate and plotting the average of the four excitations against the concentration on linear coordinate paper. Samples were run in quadruplicate and the average referred to this working curve. The standard deviation for samples containing around 10 ppm silicone additive is about 1 ppm, or a repeatability of 2 ppm (95% confidence level).

RESULTS AND DISCUSSION

A series of tests was conducted to study the storage stability of silicone dispersions prepared by different blending techniques. For the purpose of this study, the radioactive tracer technique was used to obtain quantitative information on the settling rate of the silicone. The silicones used were the 350 and 60,000 centipoise at 25 C materials prepared as mentioned in the preceding section of the paper. Ten parts per million of silicone was weighed into a shallow aluminum cup and then this was immersed in a two liter beaker containing 900 grams of dioctyl sebacate. All blends were mixed in the laboratory. Variables studied included speed, time, and temperature of mixing. The effect of prediluting the silicone with hexane or benzene (as suggested in Ref. [5]) prior to mixing into the dioctyl sebacate was also studied. Samples were stored up to 70 days in ordinary glass bottles, eighteen centimeters being the height of filling. Storage was in an air-conditioned room inside a dark cupboard to avoid thermal convection effects. Losses of silicone on glass surfaces during sampling of the bottles for counting purposes were minimized by washing all glass transfer equipment (pipettes, etc.) with 1% solution of silicone in toluene. The samples (5 ml) were taken by pipette from the top, center, and bottom

*3.00 g/l of 2,5 diphenyloxazole and 0.100 g/l of 1,4-di [2-(5-phenyloxazolyl)] benzene in toluene solution (3).

TABLE 2.—Mixing Studies Using Tritiated Silicone
(10 ppm added to dioctyl sebacate)

TEST NO.	VISCOSITY (CPS)	RPM	TIME (MIN.)	TEMP. °F.	SOLVENT	% SILICONE RETAINED AFTER SETTLING PERIOD OF: 24 DAYS	70 DAYS
A	60,000	4800	5	200	None	100	100
B	60,000	4800	5	100	None	68	52
C	60,000	4800	5	100	Hexane	80	52
D	60,000	4800	5	100	Benzene	85	—
E	60,000	670	36	200	None	80	67
F	60,000	670	36	100	None	52	48
G	60,000	670	36	100	Hexane	61	55
H	60,000	670	36	100	Benzene	54	41
I	350	4800	5	200	None	103	95
J	350	670	36	200	None	75	65
K	350	670	36	100	None	76	68

of the bottle, diluted to 50 ml with toluene, and 0.1 ml of this put in a liquid scintillator vial with 10 ml of PPO-POPOP "Cocktail." The complete results on the center samples are shown in Table 2. The silicone concentration in the top samples was not significantly different from that in the center samples. The bottom samples, in the case of unstable dispersions, always showed a high concentration of silicone.

The results of this study indicate that in order to obtain a stable suspension of silicone, considerable attention must be given to the blending technique used. For both the 350 and 60,000 centipoise materials, the conditions of high speed (4800 RPM) and elevated temperature (200 F) gave blends in which the silicone was uniformly dispersed even after 70 days of storage. These data, therefore, quantitatively confirmed the effectiveness of this technique, which had been used previously as a standard method for preparing blends in the laboratory. On the other hand, the predilution technique using hexane and benzene gave poor results as regards maintaining a uniform suspension during storage. The data, for tests A, B, C and D from Table 2, illustrate the differences noted with 60,000 cps silicone at 4800 RPM. The wide difference in results between these two techniques may be partly explained as due to the temperature effect. For practical reasons, the lower mixing temperature was used for the prediluted silicone blends in order to avoid flashing off the solvent.

The rate of shear (velocity gradient) was, as would be expected, a substantial factor affecting the stability of the dispersion. A set of experiments using the same total number of revolutions at 670 RPM rather than 4800 RPM produced substantially less stable products in all of 5 cases. This is illustrated by the data for tests A, E, B and F from Table 2, for high viscosity silicone at 200 F and 100 F blending temperatures. Similar comparisons may be made of Tests C versus G, I versus J and D versus H. The total energy input to the batch was 8.0 watt-hours in the high speed cases, as compared to 2.1 watt-hours at 670 RPM, which also tended to improve the degree of dispersion.

The effect of silicone viscosity, at both high and low speeds and 200 F, appears to be negligible. The data for tests A, I, E and J from Table 2 illustrate this point. However, at 100 F there is a distinct effect as shown by comparison of Tests F versus K. This confirms the expectations of difficult dispersion at lower temperatures (5). Microscopic examination of a blend made under these conditions revealed many droplets of 20 to 65 microns diameter. As discussed in the following section on mechanisms, this is too large for good stability.

It appears from this series of tests that, assuming some means of thinning the 60,000 cps silicone must be used, it is better to heat the entire batch than to use a solvent. It also is evident that maximum mixer speed is desirable to obtain the most stable dispersions.

MECHANISMS

It is evident, based on the observations above, that the silicone dispersions are of varying quality and that the most logical interpretation which can be made is that this variation consists primarily in size of droplet dispersion. This interpretation can be quantitatively confirmed by consideration of Stokes' law and the results of optical microscope studies. The circumstances of settling are essentially those of an unhindered situation in which the droplet fall rate will be a function of fluid viscosity, density differential, and droplet diameter. The droplet size has been estimated from the calculated settling rates for dioctyl sebacate. These calculated rates, together with those calculated for light and heavy mineral oils, are shown in Fig. 1. This is only a rough estimate but permits us to assign order of magnitude diameters to the offending droplets. The prediction is that for dioctyl sebacate-silicone dispersions, stable for 70 days, the droplets must be less than 15 microns in diameter. Therefore, it appears that the maximum limit of 100 micron droplet size (6) for optimum defoaming is still too large for good stability. In fact, microscopic examination (Fig. 2) of a stable dispersion similar to Test A indicated no droplets of over 3 microns diameter to be present, while examination (Fig. 3) of a sample similar to the unstable dispersion from Test F revealed droplets up to 110 microns. These micrographs were made at 1000 ppm, as the normal 10 ppm gave too few particles in the field to permit a representative picture.

Additional information on the size of the smaller droplets was obtained from samples of the radioactive dispersion from Tests A and C (Table 2) which were filtered (after shaking) through a Millipore filter (grade HA, 0.45 micron diameter), the filtrate then being examined for radioactivity. The filtrate showed 1.7 and 1.3 parts per million of silicone respectively, indicating that essentially all of the silicone in the emulsified form had been removed. This remainder is of the same order of magnitude as the solubility of silicone in oils (1), (6) and (7). It therefore appears that few, if any, droplets are below 0.45 microns in diameter.

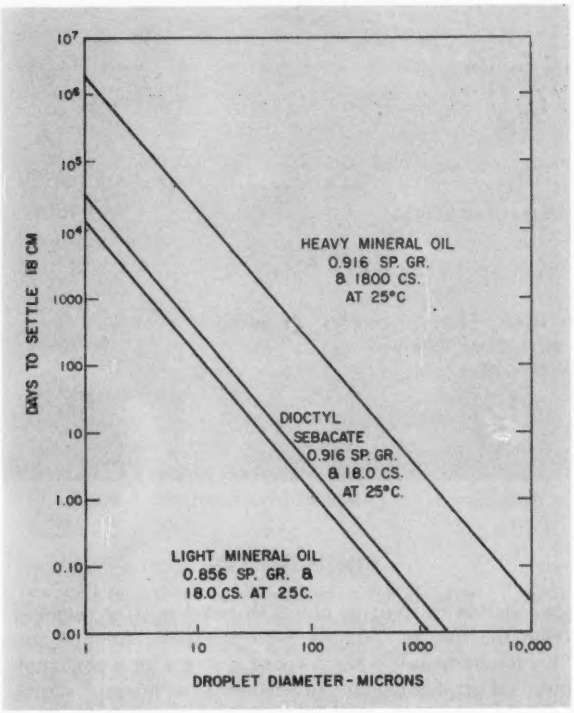

Fig. 1. Calculated Time of Fall of Silicone Droplets (0.976 gm/cc @ 25 C).

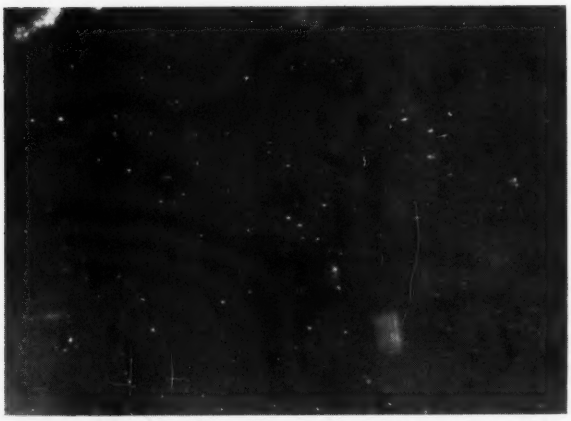

Fig. 2. 1000 PPM Dispersion, Made at 200 F and 4800 RPM.—(65 ×)

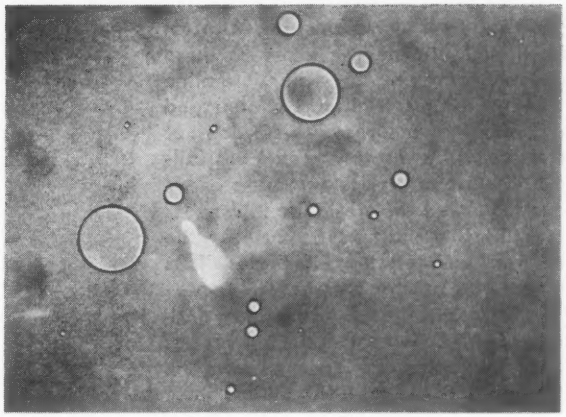

Fig. 3. 1000 PPM Dispersion, Made at 100 F and 670 RPM.—(65 ×)

CONCLUSIONS

It may be concluded from this study that a stable dispersion of silicone in a non-aqueous fluid requires a droplet size in the 10 micron range, and that this can best be achieved by a moderately high degree of heat and agitation. The solvent technique tested was definitely inferior. The studies, while limited to one fluid, indicate that specific laboratory procedures can be worked out as readily for other fluids.

It may also be concluded that two adequate methods have been developed for the study of settling of silicone from non-aqueous fluids. The choice between these two methods will depend on the circumstances, the radioactive tracer method being by far the more sensitive and accurate, but also the less widely applicable. The emission technique has the advantage of being usable under all conditions.

ACKNOWLEDGMENT

We wish to thank C. R. Hodgkins, who was instrumental in developing the emission procedure described, for his very important contributions to the solution of this part of the problem.

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Residual Stress vs. CUTTING FLUID SELECTION *When Grinding High Temperature Alloys*

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Residual stresses are inherent in practically all operations where metal is shaped. The degree to which these stresses are present will determine whether the resulting part is damaged by either measurable distortion or stresses which result in premature failure. The use of high temperature alloys has increased the need for low stress machining operations because of the inherent susceptibility of these alloys to induced residual stress.

Stress produced during the grinding of high temperature alloys has been evaluated by test methods. The method uses the resulting deflection of ground test specimens as the basis for evaluation. Results are measurable in terms of both stress level (pounds per square inch) and depth of stress penetration.

Grinding fluids are among the variables which have a substantial effect on stress produced. This paper presents the effects of grinding fluids on induced stress for both basic types, oil vs. water and for the various chemical analyses of oils.

These tests show a substantial reduction in stress for sulphurated oils as compared to water based fluids.

Space age uses have demanded alloys capable of withstanding high stress at elevated temperatures. In addition, these metals must be used close to the design limits in order to reduce weight to an absolute minimum. These factors result in highly stressed and often thin walled parts that are subject to damage from residual stress.

Residual stress, for practical purposes, can be defined as a stress which remains in the surface of a metal part after mechanical restraint has been removed. This stress results from the superficial displacement of metal and the subsequent attempt of this metal to return to its original position. The resulting stress can be either tensile or compressive, depending on the action that causes it. As a general rule, mechanical action causes compressive stresses, while tensile stresses are the result of thermal upsetting.

Shop experience has proved high temperature alloys to be extremely susceptable to residual stress during machining operations, and particularly so in grinding. This stress sometimes is apparent in parts that visibly warp or go out of dimension after removal from the fixture. A more serious stress effect is the "hidden damage" in the form of a stress on a part that does not bend or change dimension. The "unseen" stress can cause failure under load while the part is in operation with a resulting damage to parts other than itself. It is fortunate when an alloy visibly cracks under a high residual stress when in process rather than in operation.

To determine the extent and nature of residual stress, it was necessary to perform a laboratory study. In the laboratory, the effect of grinding variables could be solated and studied on expendable specimens. Once the variables and their effects were established, measures were taken to minimize the induced stress. One of the variables which had a great effect on residual stress was the grinding fluid used. This paper emphasizes the results obtained in comparisons of both fluid base (oil vs. water) and chemical additions to oil.

TEST PROCEDURE

The basic technique used to analyze the effect of various grinding fluids on the residual stress level was to grind flat strips of metal and measure the change in flatness (deflection) of the part after the completion of the grinding.

Presented at the 15th ASLE Annual Meeting in Cincinnati, Ohio, April, 1960

EQUIPMENT USED

1. Surface grinder, size 6 x 18, equipped with infinitely variable spindle speed.
2. Special fixture to hold the flat strips for grinding.
3. Special gage to measure deflection of the grinding sample.
4. Strain gage dynamometer to measure horizontal and vertical grinding force.

MATERIALS

The materials selected for analysis are typical of those used in jet engine manufacture and may be generally classed as high temperature alloys. Table 2 lists these alloys. After the samples had been machined and ground to size (6 in. long x ¾ in. wide x 0.200 in. thick), they were mechanically straightened. They were then heat treated in a "supported" position to provide a "stress free' sample with minimum distortion.

GRINDING PROCEDURE

A complete description of the grinding procedure used in these tests is described by Halverstadt (3) in the bibliography, but the following steps will describe the basic technique used in the grinding operation.

1. Measure and record the initial deflection of the sample.
2. Dress the grinding wheel.
3. Mount the test sample in the grinding fixture and grind to 0.120 in. thick using predetermined grinding conditions.
4. Measure the sample for change in deflection.
5. Dress the grinding wheel.
6. Grind the specimen to 0.100 in. thick.
7. Make final change in deflection measurements of the specimen.

MEASUREMENTS

The equations and general method used to make the stress analysis is that developed by Frisch and Thomas (1) and further described by Halverstadt (3). This technique is based on the fact that a residual stress in the surface of a specimen will cause the sample to bend. If the sample bends in a convex manner, the net residual stress is compressive. If the surface ground bends in a concave manner, the net residual stress is tensile in nature. By etching away successive layers of the stressed surface and measuring the resulting change in deflection, basic data for the stress equations is obtained.

DISCUSSION

Types of cutting fluid can be classified in many ways, but for the purposes of this paper, they will be classed as: oil based or water based. There are obviously broad categories, as the water based can be chemical or soluble oil while numerous additives can be made to mineral oil. The first part of this discussion compares water vs. oil while the second section deals with oils and their additives.

Oil vs. Water Based

Alloys designed for the higher temperature ranges react similarly to changes in cutting fluid whether they are iron based or nickel based. The fluid effects cannot be completely separated from the effects of such variables as wheel speed, wheel grade, and feed rates. Use of the proper fluid cannot counteract the detrimental effect of high wheel speed or of excessively hard wheels. However, when job factors will not permit ideal grinding conditions, the use of the correct fluid will balance to a great extent any detrimental effect of a compromise on other conditions.

The effect of grinding fluid for high and low wheel speed ranges is shown in Figs. 1 and 2. At high speed, there is a high level of stress produced by both types of fluid, but the oil indicates a compressive overlay on the high tensile stress. While this combined stress is not desirable, it is preferable to the pure tensile stress of the water based fluid. At low speeds, the effect of oil

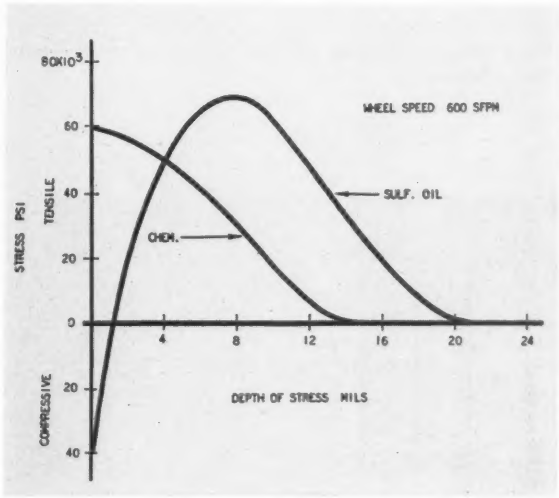

Fig. 1. Effect of Grinding Fluid on Residual Stress at High Wheel Speed.

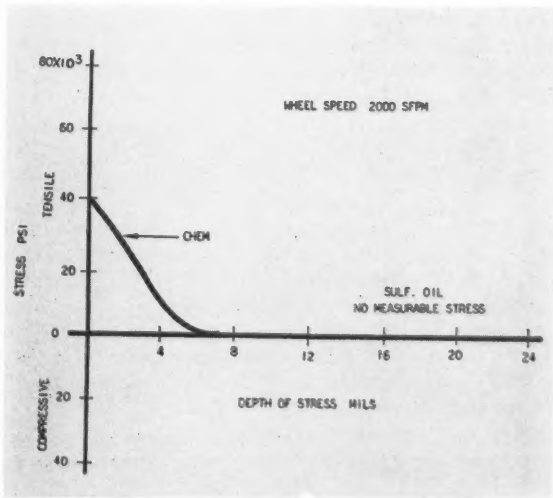

Fig. 2. Effect of Grinding Fluid on Residual Stress at Low Wheel Speed.

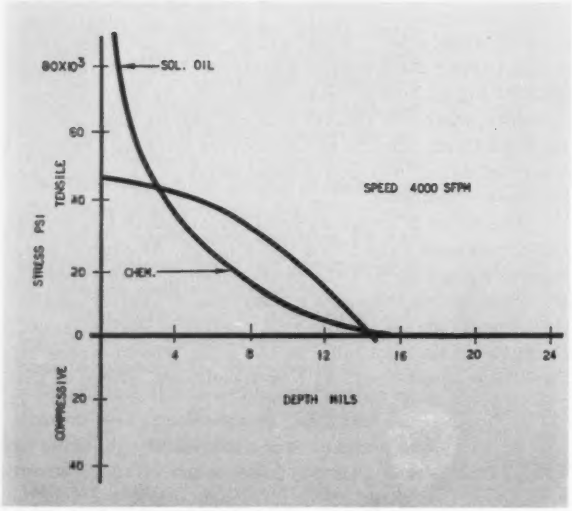

Fig. 3. Effect of Grinding Fluid on Residual Stress at Medium Wheel Speed.

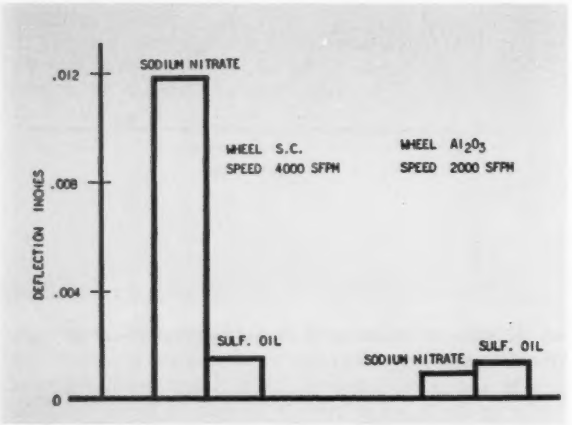

Fig. 4. Effect of Grinding Fluid on Residual Stresses When Grinding Titanium.

over water is pronounced as shown in Fig. 2. Fig. 3 is presented to compare the effects of chemical vs. soluble oil fluids. As is typical of water based fluids in grinding high temperature alloys, there is a high tensile stress produced. This comparison indicates that soluble oils under these test conditions produce higher surface stresses than do chemical emulsions.

A somewhat different result occurs for a comparison of oil vs. water based fluids on titanium alloys. Titanium is not a true high temperature alloy compared to nickel based alloys, but its strength to weight ratios make it ideal for use in conjunction with super alloys. There are two techniques for grinding titanium that are generally used: aluminum oxide wheels and low wheel speeds; silicon carbide wheels and medium wheel speeds. A highly sulphurated oil and a sodium nitrite water mixture were tested using both grinding wheel materials. Fig. 4 shows a comparison of the deflection obtained with the two fluids under two specific conditions. The following recommendations are made based on Fig. 4 and other unpublished data.

1. Silicon carbide wheels and sulphurated oil are preferred because they permit higher wheel speeds with low stress deflection than do aluminum oxide wheels with either oil or water fluid.
2. Oil has one major drawback, in that fire hazards are inherent when grinding titanium.
3. The sulphurated oil used should not contain chlorine as there is a danger of surface cracking during subsequent heat treatment if chlorine has been absorbed.

Grinding Oil Comparisons

It has been shown that correct grinding fluid selection has a great effect on the level of residual stress. It has also been shown that in high temperature alloys, sulphurated oil is significantly effective in reducing the level of residual stress. Since only one oil had been used in the previously discussed tests, a special test was conducted to compare the effects of twelve (12) commercial grinding oils. Table 1 and Figs. 5, 6 and 7 show data for these tests.

TABLE 1.—Oil Analysis

OIL	FLASH POINT (°F)	FIRE (°F)	VISCOSITY @ 100 F	SUPONIFICATION NO.	% TOTAL SUL.	% ACTIVE SUL.	% TOTAL CL.	% ACTIVE CL.
A	350	395	185-200	65	3.3	2.1	1.0	—
B	250		100-105	25	.8	—	5.0	—
C	300	340	200	10	1.3	.75	.5	.25
D	395	445	252		3.19	3.02	—	—
E	375	415	138	16.6	.5	.3	.09	.09
F	350	395	180-200	50	3.2-3.6	2.1	2.0	1.8
G				76.5	2.3	—	—	—
H	345	390	280-300	60	—	—	—	—
I	290	—	110-125	10.1	2.64	1.51	.81	.41
J	310	—	160-180	5.5	1.16	.40	.90	—
K			195	52	2.6	—	—	—
L	345	390	180-195	25	.9	—	—	—

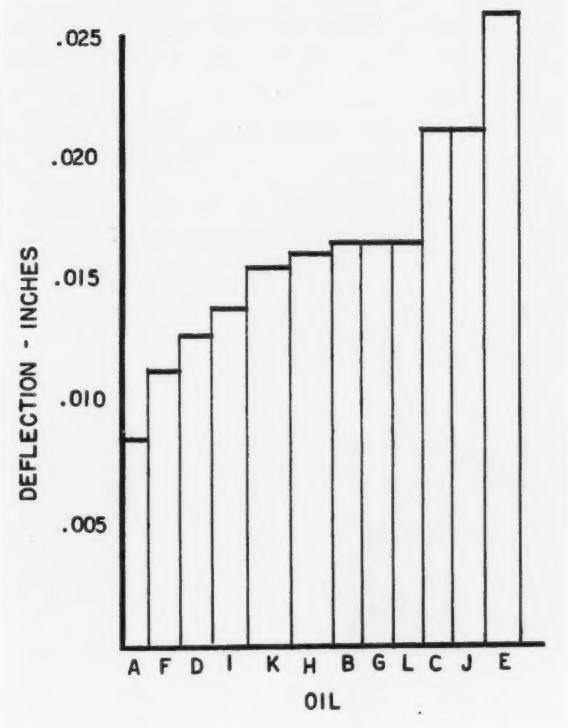

Fig. 5. Effect of Type of Grinding Oil on Residual Stress Under Rough Grinding Conditions.

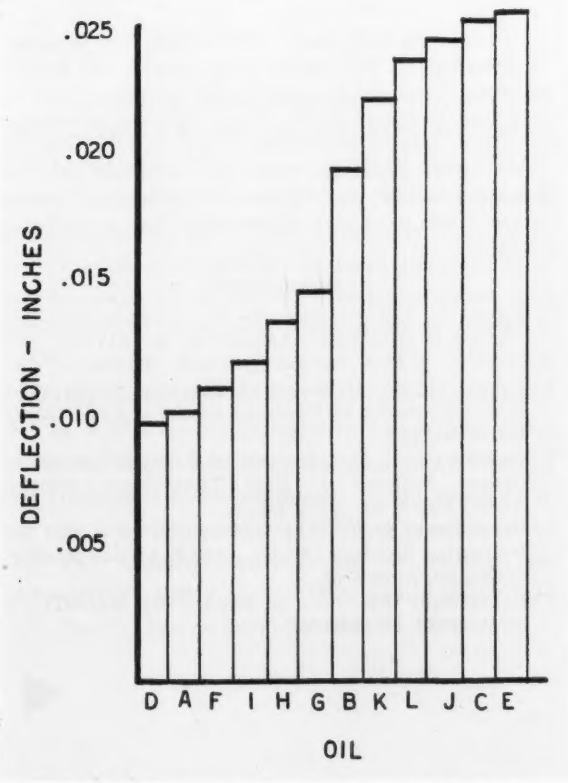

Fig. 6. Effect of Type of Grinding Oil on Residual Stress Under Finish Grinding Conditions.

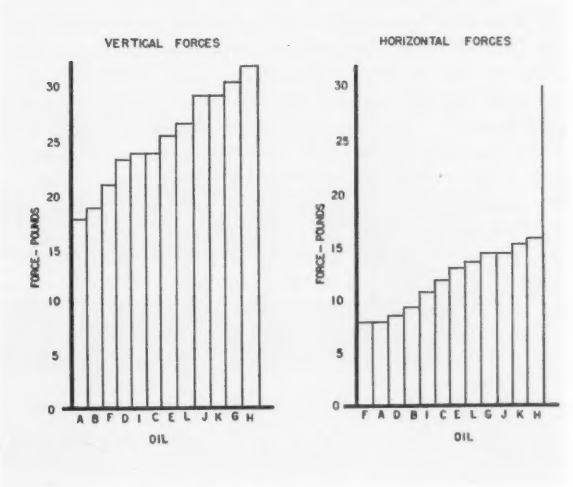

Fig. 7. Effect of Type of Grinding Oil on Grinding Forces.

Because it was desirable to limit the variables to those of the grinding fluid only, a special set of grinding conditions was used. They were:

Material	Alloy 1., Table 2 @ R_c 30.
Wheel Type	(Friable) A60K6V
Wheel Speed	4000 SFPM
Table Speed	15 SFPM
Type of Grind	Plunge Cut
Down Feed	Rough grind: 0.002 in./pass
	Finish grind: 0.001 in./pass

The above grinding conditions were considered rather harsh, but this was desirable in order to obtain more pronounced differences in residual stress.

Figs. 5 and 6 show the comparison of deflection after rough and finish grinding. Deflection after rough grinding showed a different pattern than for finish grinding for the four oils giving the lowest deflection levels. However, the significant factor is that, in both cases, the four low deflection oils are the same ones in a slightly different order. A high *active* sulphur content appears to be very effective in reducing tensile residual stress. Oil D, which had the highest active sulphur content, showed the least deflection after the finish grind. Deflection after the finish grind was considered to be the best measure of residual tensile stress because the samples were approximately 20% thinner after finish grinding than after rough grinding and, therefore, more sensitive to deflection.

It appears that a high percentage of chlorine has little effect on the level of residual stress. Oil B, which had the highest chlorine content, was rated 7th on both rough and finish grinding. However, tests showed that Oil B produced relatively low grinding forces. This indicates that in the absence of a high active sulphur content, a high chlorine content may offer some benefit.

With the exception of Oil B, the oils which showed the most favorable results on deflection measurements also proved to be the most favorable with respect to grinding forces. Fig. 7 lists force data. This pattern was repeated in the grinding ratio measurements which were made but not reported in this paper. An analysis of the grinding process indicates that an oil which gives both a favorable "G" ratio and a low deflection helps the cutting process by reducing grinding forces in the following manner.

The individual grinding wheel grits stay sharp longer. Therefore, the wheel has fewer cycles of loading and breakdown. The sharper wheel not only cuts cooler, resulting in less deflection, but less energy is required than to cut with a dull grinding grit. This reduced amount of energy results in decreased grinding forces.

In a later test series, two additional oils were tested for the purpose of corroborating the results of the above tests. The oils were chosen solely on a basis of *active sulphur* content. The results of this test proved that the original test results were valid and that a satisfactory grinding oil for use on high temperature alloys *must* contain a high active sulphur content.

TABLE 2.—Alloy Composition

ALLOY DESIGNATION	NOMINAL COMPOSITION, PERCENT									
	CO	CR	MN	NI	MO	FE	TI	AL	C	OTHER
Iron Base										
1. Fe-Ni-Cr,	286	15	1.5	25	1.3		2.1	.35	.08	.007 Bo
Titanium										
2. Ti-Al-Mo				.07	4		89	7		
Nickel Base										
3. Ni-Cr-Co, 500	19.5	19		47	4	4	2.9	2.9	.08	
4. Ni-Cr-Co, 252	10	19		55	10	2	2.5	1	.15	

CONCLUSIONS

The over-all conclusion from these tests is that sulphurated oils are superior to water based fluids as a grinding medium for super alloys. This paper does not attempt to explain the action by which sulphur accomplishes its results. Indications are that a favorable reaction takes place between the metal being cut and the sulphur when they are subjected to the heat of cutting. This, evidently, provides a form of lubrication which lowers heat and force and, thereby, lowers induced stress.

From the tests made to compare grinding oil compositions, three factors are evident:

1. The data indicates that there is good correlation between "G" ratio, tensile stress, and grinding forces. From this correlation, it can be expected that a grinding oil which results in a low residual tensile stress will also give a favorable grinding ratio, and low grinding forces.
2. The most important constituent of a grinding oil is the active sulphur content. Total sulphur content means little unless it reflects the amount of active sulphur. The minimum amount of active sulphur in oils for grinding high temperature alloys should be above 1.5%.
3. The chlorine content in a grinding oil seems to provide little benefits with respect to "G" ratio, deflection, and grinding forces. In an oil that contains little or no sulphur, the addition of a large amount of chlorine seems to have some effects.

Laboratory tests such as these bring out interesting information, but, unless their results can be put to practical use, little value is gained. The results of these tests have had practical application in jet engine production. Qualitative results have shown that the use of low stress grinding conditions, including the use of sulphurated oil, will eliminate cracking and stress failure when they are substituted for undesirable conditions.

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Operating Characteristics of 75 millimeter bore

BALL BEARINGS

At Minimum Oil Flow Rates Over a Temperature Range To 500 F

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Experiments were performed to determine the actual minimum oil flow requirements of deep groove ball bearings at DN values of 6.0×10^5 to 9.75×10^5, thrust loads of 1000 and 3000 pounds and bearing temperatures to 500°F with air-oil mist lubrication. With a synthetic diester oil, required minimum flows (on the order of 10^{-5} to 10^{-2} lb/min) increased with increasing DN, load, and temperature to 400°F. At 500°F varnish deposits formed which apparently aided lubrication and reduced the flow required below that at 400°F. Bearing temperature and torque generally decreased with decreasing oil flows down to the minimum-oil-flow condtions. As an example of the very low oil flows required, it was possible to run size 215 bearings satisfactorily at 12,000 rpm (9.0×10^5DN), 400°F, and a 1000 pound thrust load with flows as low as 0.003 pounds per minute.

Presented at the 15th ASLE Annual Meeting in Cincinnati, Ohio, April, 1960.

INTRODUCTION

There are bearing applications, such as missiles and space vehicles, in which a once-through lubrication system operating with minimum possible oil flow could advantageously be employed. Weight reduction of the system might be realized, especially if a pressurized gas supply is available in the vehicle. To intelligently design a once-through lubrication system, the oil-flow requirements of the bearings must be known. This information is also necessary to determine whether a once-through system has any advantage over a recirculating system.

Very little experimental information is available in the literature on the minimum-oil-flow requirements of rolling-contact bearings under steady-flow conditions. Jones and Wilcock (1) and Booser and Wilcock (2) report some data on the relation of the running time to failure as a function of the quantity of oil applied for 30- to 50-millimeter-bore ball bearings when given an application of oil prior to test. Booser and Wilcock (2) deduce the minimum oil quantity required on a steady-flow basis from the analysis of their data; this information was obtained to secure quantitative estimates of the required bleed rate of greases. Boyd and Eklund (3) report some data on minimum oil requirements of 25-millimeter-bore ball bearings at 9×10^5 DN (product of bearing bore in millimeter and speed in rpm), and temperatures of ball and roller bearings at low oil flows.

The object of the investigation reported herein was to obtain quantitative data on the actual oil-flow requirements of ball bearings at various speeds, loads, and temperatures. The results of some preliminary experiments to determine the minimum-oil-flow requirements of 75-millimeter-bore deep-groove ball bearings are reported. Tests were conducted at DN values of 6×10^5 to 12×10^5, thrust loads of 1000 and 3000 pounds, and temperatures to 500° F. In all the tests the synthetic diester lubricant was supplied to the test bearings in the form of an air-oil mist.

APPARATUS

A schematic diagram of the test bearing arrangement is shown in Fig. 1. With the exception of the test bearing arrangement, the rig is basically the same as that used by Anderson, Macks, and Nemeth (4). The two test bearings were loaded in thrust by a flexible stainless-steel bellows which was pressurized with oil (Fig. 1). The bellows was calibrated on a tension-compression test machine, and the load during test was determined from the pressure-load calibration curve. The load capacity of the system was about 5000 pounds. The available speed range of the test shaft was 1,100 to 36,000 rpm, controllable to within ±50 rpm at all speeds.

Electric resistance heaters were wound in a recess in each bearing housing after which the recesses were filled with cast aluminum. Six iron-constantan thermocouples were located in each test bearing outer-race housing at 60° intervals around the outer-race periphery at the axial mid-points of the bearings. These thermocouples were embedded in the housing flush with the housing inside diameter. One of the thermocouples was connected to the heater control instrument which maintained the desired test bearing outer-race temperature.

A strain-gage beam connected to the bearing housing (Fig. 1) was used to measure test bearing friction torque. The strain-gage output was amplified and recorded on a photoelectric potentiometer. Torque could be measured with an average sensitivity of 0.05 pound inch.

In order to obtain the range of oil flows required for the investigation, several lubrication systems for the test bearings were devised. Two commercial air-oil mist lubricators were calibrated with 0.089-inch-diameter, 0.56-inch-long reclassifiers recommended by the manufacturer.

These units were calibrated at various supply pressures using test tubes partially filled with glass fiber wadding to trap the flow. The highest flow obtainable with the larger of the two commercial units was 0.003 pound per minute, while the lowest solid oil flow obtainable with a practical nozzle (0.021-in. diam.) was 0.065 pound per minute. To bridge this gap a third air-oil mist generating system, called the experimental air-oil mist lubricator, was designed. Sketches of this system and the reclassifiers used with the commercial lubricators are shown in Fig. 2. The experimental lubricator employed a pressurized tank and a capillary to feed lubricant into the flow line downstream of a Venturi. Fig. 3 summarizes the oil-flow ranges obtainable with the three air-oil mist systems employed. The flows of the three air-oil mist systems overlap so that certain oil flows could be obtained with two systems. When incipient failures occurred in these overlapping regions, the test was repeated with the alternate lubrication system, with good reproducibility of results.

All tests were performed with size 215 deep-groove M-50 tool-steel bearings with two-piece inner-race-riding, silver-plated iron-silicon-bronze retainers. M-50 tool steel has the following composition: 0.80 percent carbon; 4.00 percent chromium; 1.10 percent vanadium; 4.00 percent molybdenum; balance iron. The test bearings had an average radial clearance of 0.0004 inch, with a range of radial clearances from 0.0002 to 0.0006 inch.

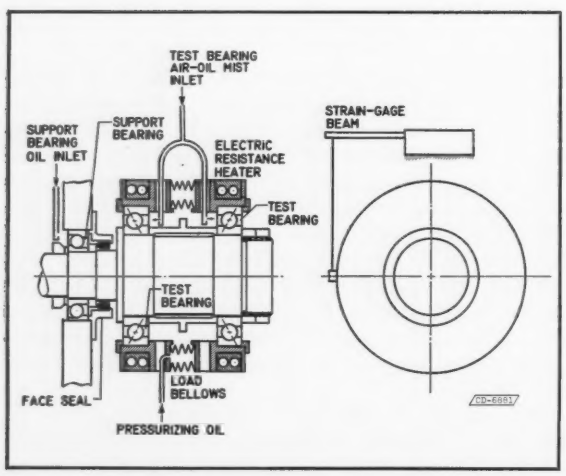

Fig. 1. Test bearing arrangement.

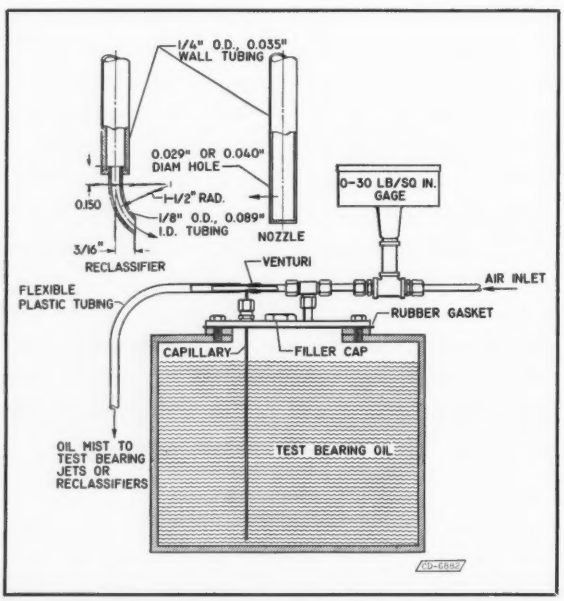

Fig. 2. Experimental air-oil mist test bearing lubrication system.

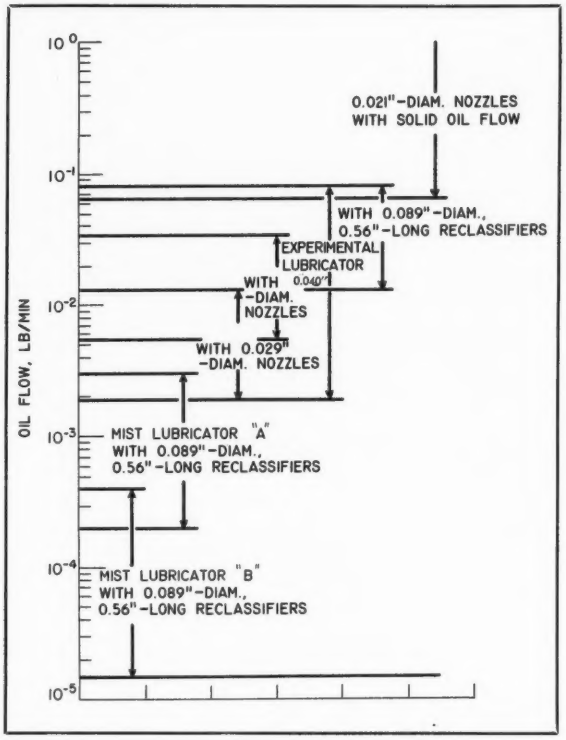

Fig. 3. Minimum-oil-flow lubrication systems' flow ranges.

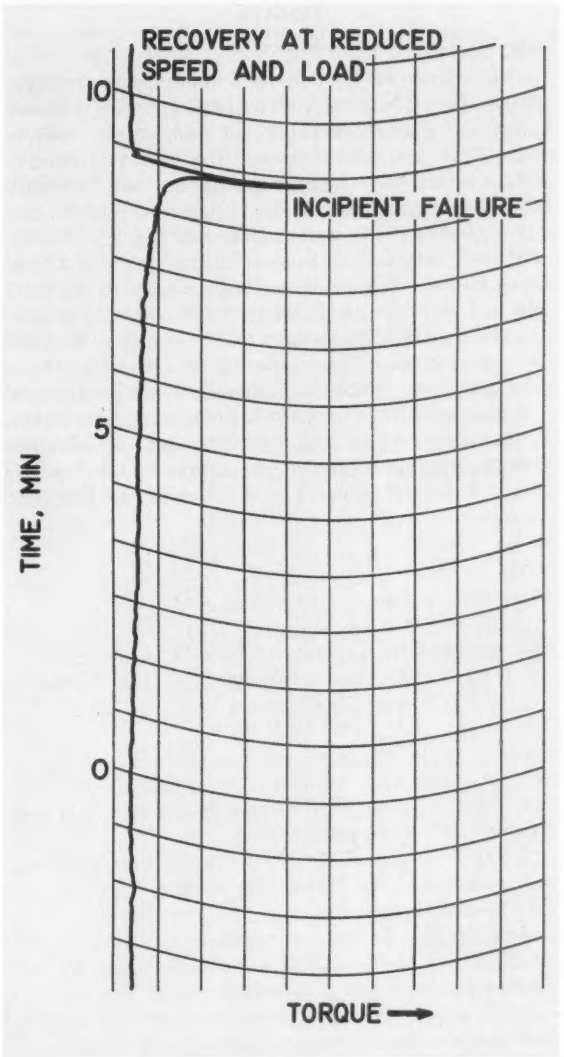

Fig. 4. Typical bearing torque curve at incipient failure. DN, 6×10^5; load, 1000 pounds (thrust); temperature, 400 F; flow per bearing, 5×10^{-4} pound per minute.

PROCEDURE

Minimum-oil-flow data points were obtained in the following manner. After the desired load, speed, and temperature were selected, a flow from previous experience known to be sufficiently high to ensure proper bearing operation under the chosen conditions was set. The test bearings were run at these conditions for 3 hours to ensure that equilibrium conditions prevailed. The flow was then approximately halved, all other conditions remaining the same, and another 3-hour run was attempted. This procedure continued until an indication of incipient failure occurred. This indication consisted of a sudden sharp rise in test bearing torque (Fig. 4), indicating momentary inadequate lubrication due to a local breakdown of the boundary lubricating film without significant surface damage. The speed and load were then immediately lowered, and the bearing was run at an idling speed for approximately ½ hour to allow bearing recovery, as indicated by lower and stable bearing torque and temperature. In a few tests early in the program the 3-hour run was repeated with twice the flow at which incipient failure was indicated. No failures occurred in these repeat runs, which indicated that the damage suffered by the test bearings during the incipient failures was negligible.

The DN range of this investigation was limited by the range of oil flows obtainable with the three air-oil mist lubrication systems. It was not possible to obtain oil flows low enough to produce failures at DN values below 6×10^5, nor was it possible to obtain oil flows high enough to prevent failures at DN values above 9.75×10^5.

Two basic types of tests were conducted. Twenty tests were run with no external heat flow to the bearings. In these tests the bearing temperature was allowed to seek its equilibrium level, and was a function of both DN and load. Thirty four tests were run with the test bearing outer-race temperature maintained approximately constant by varying the heat flow into the bearing housings. Tests of this type were run at bearing temperatures of 300°, 400°, and 500° F.

RESULTS

Tests Without Heat Addition

The effect of DN on minimum-oil-flow requirements at thrust loads of 1000 and 3000 pounds is shown in Fig. 5. The repeatability of test results can be gauged from the scatter at each DN value. Minimum-oil-flow requirements increase with DN and also with load. The flow required at the higher load is about two orders of magnitude higher than that required at the lower load. The best-fit lines at the two loads, obtained by the method of least squares, are shown as the solid lines in Fig. 5. To facilitate correlating all these data, a slope based on the average of the slopes at the two loads was chosen. The average slope lines are drawn as dashed lines. Since the dashed lines are parallel and in close proximity to the best-fit lines, an approximation of minimum-oil-flow requirements can be obtained from the empirical relation involving the oil flow, the load, and the DN value. The equation of one line is of the form

$$\frac{\log Q_2 - \log Q_1}{DN_2 - DN_1} = \eta \qquad [1]$$

where Q is oil flow, η is the slope, and

$$Q_2 = Q_1 10^{\eta(DN2-DN1)} \qquad [2]$$

At a specific DN, $Q = f(W)$ where W is load.

If $Q = CW^K$, the constants η, C, and K can be obtained. The resulting relation is

$$Q = (1.51 \times 10^{-18})\ (W^{4.22} 10^{1.18(S-8)}) \qquad [3]$$

where Q is the minimum oil flow below which failure would be imminent (lb/min), W is thrust load (lb), and S is DN value$\times 10^{-5}$ (done to avoid using large numbers in the calculations).

The degree of correlation for the lines in Fig. 5 was quite good, the correlation coefficients being 0.93 for the 1000 pound line and 0.91 for the 3000 pound line. Since the correlation coefficient is a statistic that measures the usefulness of a regression line for estimating purposes and a correlation coefficient of 1.0 is considered perfect, the values of 0.93 and 0.91 obtained here make estimation quite accurate and reliable.

When a bearing is operated at a constant load without external heat addition, its temperature rises with increasing DN value. Therefore, part of the effect of DN on minimum-oil-flow requirements may be indirectly due to temperature rise. As a preliminary check on the importance of temperature, the data of Fig. 5 are shown in Fig. 6 with log minimum oil flow plotted against average bearing outer-race temperature. The correlation of minimum oil flow with temperature is about as good as with DN, indicating that temperature may overshadow DN value as the important variable. Correlation coefficients of 0.99 and 0.87 were obtained for the 1000 and 3000 pound lines of Fig. 6.

An empirical relation between minimum oil flow, temperature, and load similar to that of Equation [3] was derived and is as follows:

$$T = 53.5\ (\log Q + 2.38 \log W + 1.18)$$

where

T temperature, °F
Q minimum oil flow below which failure would be imminent, lb/min
W thrust load, lb

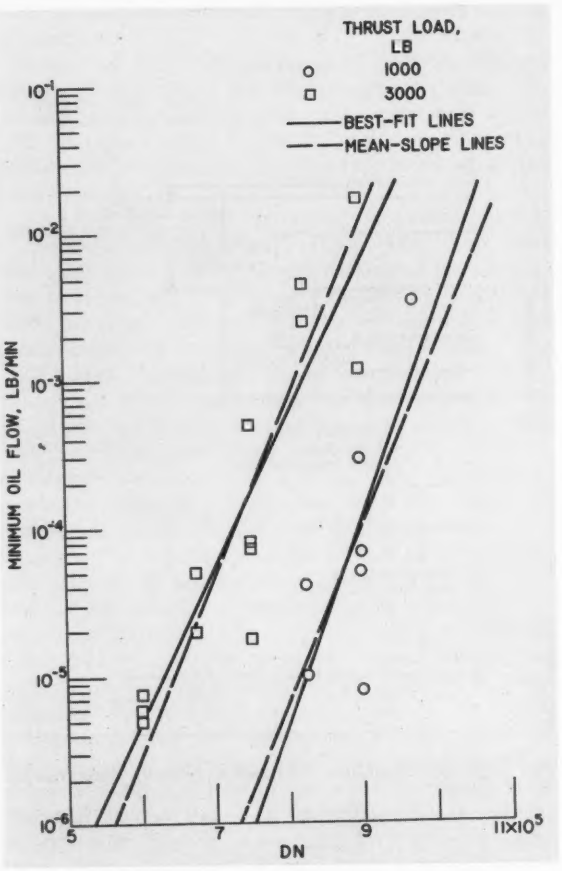

Fig. 5. Minimum-oil-flow requirements of size 215 deep-groove ball bearings as function of DN at two thrust loads. No external heat added to bearings.

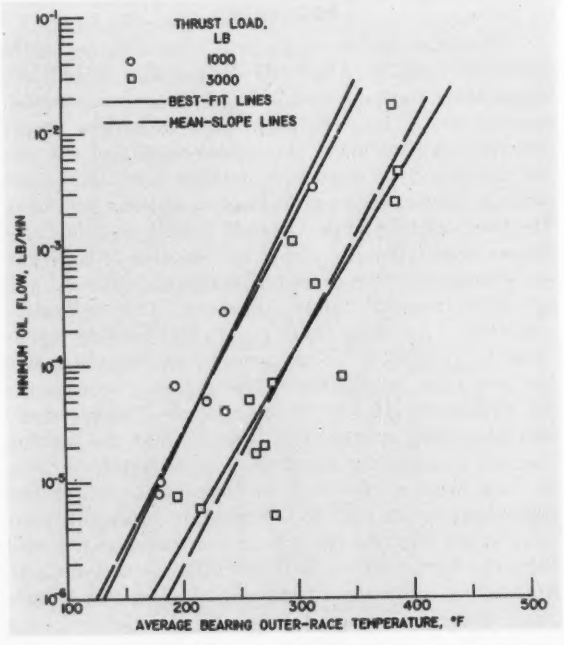

Fig. 6. Minimum-oil-flow requirements of size 215 deep-groove ball bearings as function of bearing temperature at two thrust loads. No external heat added to bearings.

With a known value of thrust load and DN, Q can be calculated from Equation [3] and an expected bearing temperature can then be obtained from Equation [4]. It must be remembered that bearing temperature depends on the heat-transfer characteristics of the bearing environment. Also, Equation [4] would not apply to an installation with a different size bearing. Therefore, Equation [4] is true only for this particular installation. However, the constants in Equation [4] could be determined for any particular installation.

These relations are true for the bearing and lubricant combinations employed in this study with an air-oil mist lubrication system. They cannot be extrapolated for use with other bearing sizes or lubricants or with solid oil jet lubrication, but it would be interesting to see how these relations differ with other lubricants and bearing sizes.

Fig. 7 shows plots of bearing temperature against oil flow for both the air-oil mist lubrication system used to compile these data and for a solid oil lubrication system. It is interesting to note that, as oil flow is decreased with solid oil lubrication (Fig. 7[a]), the bearing temperature increases, whereas the opposite is true with an air-oil mist system (Fig. 7[b]). This effect has also been reported by Boyd and Eklund (3), Accinelli and Beaubien (5), and Getzlaff (6). A possible explanation of this fact is that, under minimum-oil-flow conditions, thin oil films are involved and less heat is therefore generated by oil shearing than would be the case at higher flows. Thus, reduced bearing temperatures would be obtained. The increase in bearing temperature with increasing oil flow in the minimum-oil-flow range (Fig. 7[b]) implies that the rate of heat generation in the bearing due to shearing of the oil increases at a faster rate than does the added cooling capacity. The data of Fig. 7(a) and (b) indicate that this situation will prevail as oil flow is increased until a peak bearing temperature is reached. Further increases in oil flow result in diminishing temperatures because cooling capacity is now increasing at a faster rate than is the shear work on the oil. Getzlaff (6) presents some plots which illustrate this effect.

Fig. 8 further illustrates that as oil flow is decreased with air-oil mist lubrication, the bearing temperature decreases. This is true of bearing torque also as shown in Fig. 9. As flow is decreased, torque decreases because of reduced oil shearing. Data for some load and speed conditions (i.e., 9×10^5 DN and 3000 lb load on Fig. 8, and 7.5×10^5 DN and 3000 lb load on Fig. 9) are shown for more than one bearing pair to illustrate reproducibility.

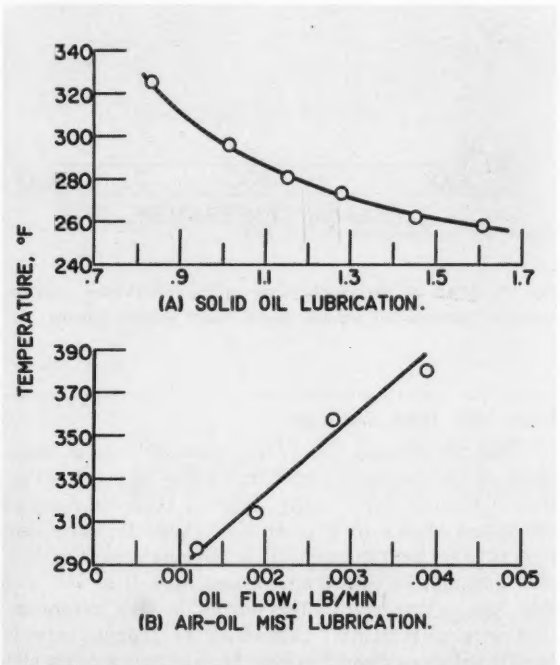

Fig. 7. Bearing temperature as function of oil flow for size 215 deep-groove bearings with solid oil lubrication and air-oil mist lubrication. Speed, 12,000 rpm; load, 3000 pounds (thrust); lubricant, MIL-L-7808 synthetic oil.

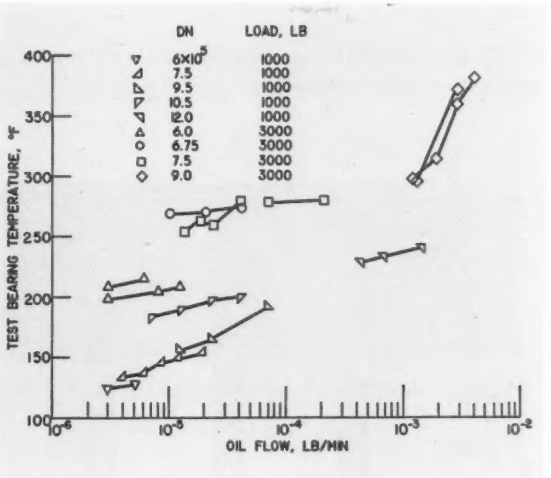

Fig. 8. Test bearing temperature as function of oil flow at various values of load and DN for air-oil mist lubrication.

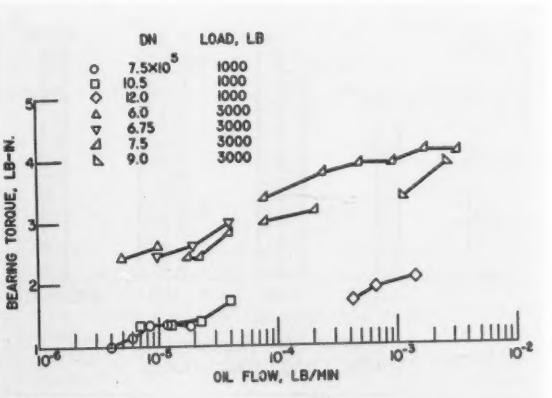

Fig. 9. Test bearing torque as function of oil flow at various values of load and DN for air-oil mist lubrication.

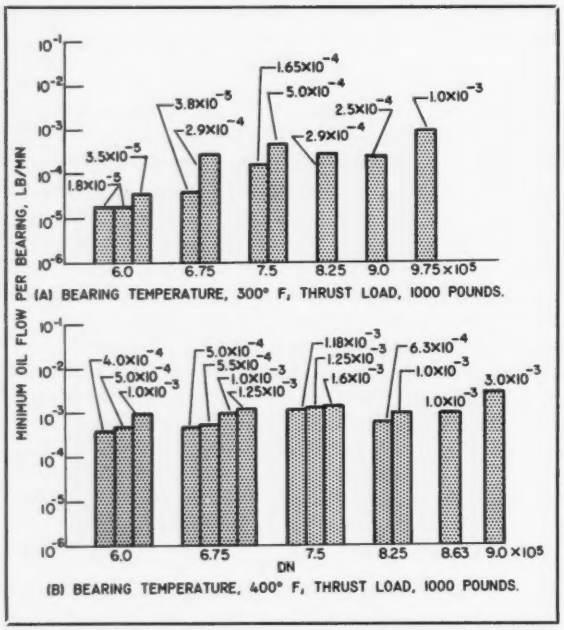
MINIMUM OIL FLOW PER BEARING, LB/MIN
(A) BEARING TEMPERATURE, 300° F, THRUST LOAD, 1000 POUNDS.
(B) BEARING TEMPERATURE, 400° F, THRUST LOAD, 1000 POUNDS.
DN

Fig. 10. Effect of DN on minimum-oil-flow requirements at constant bearing temperature.

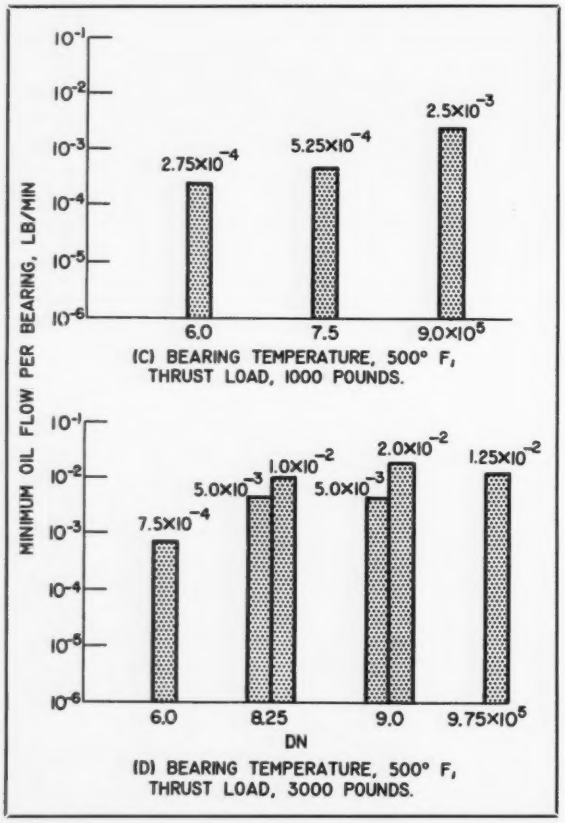
MINIMUM OIL FLOW PER BEARING, LB/MIN
(C) BEARING TEMPERATURE, 500° F, THRUST LOAD, 1000 POUNDS.
(D) BEARING TEMPERATURE, 500° F, THRUST LOAD, 3000 POUNDS.
DN

Fig. 10. Concluded. Effect of DN on minimum-oil-flow requirements at constant bearing temperature.

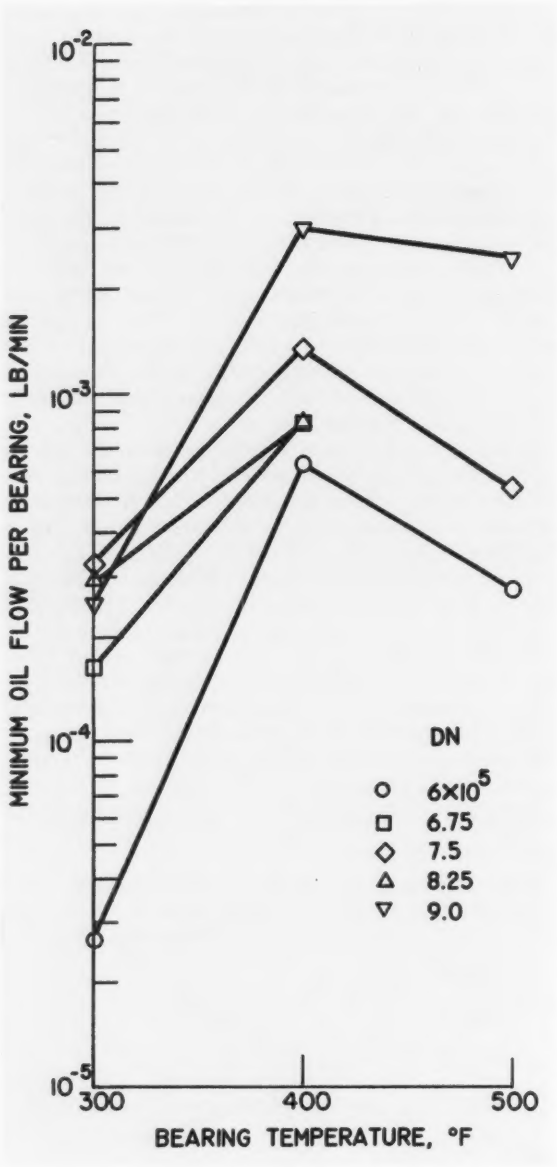

Fig. 11. Effect of temperature on minimum-oil-flow requirements at various DN values. Load, 1000 pounds (thrust).

Tests with Heat Addition

Fig. 10(a) and (b) reveal that DN has a lesser effect on the required minimum oil flow at 400 F (Fig. 10[b]) than at 300 F (Fig. 10[a]). Data obtained at 500 F are shown in Fig. 10 (c) (1000 lb load) and Fig. 10(d) (3000 lb load). It is obvious from the data of Fig. 6 and 10 that temperature, as well as DN and load, has an important effect on the bearing minimum-oil-flow requirements. The effect of temperature is illustrated more clearly in Fig. 11, which is a cross plot of the data of Fig. 10(a) to (c). At a constant DN, minimum-oil-flow requirements increase sharply in going from 300 to 400 F but actually decrease in going from 400 to 500 F.

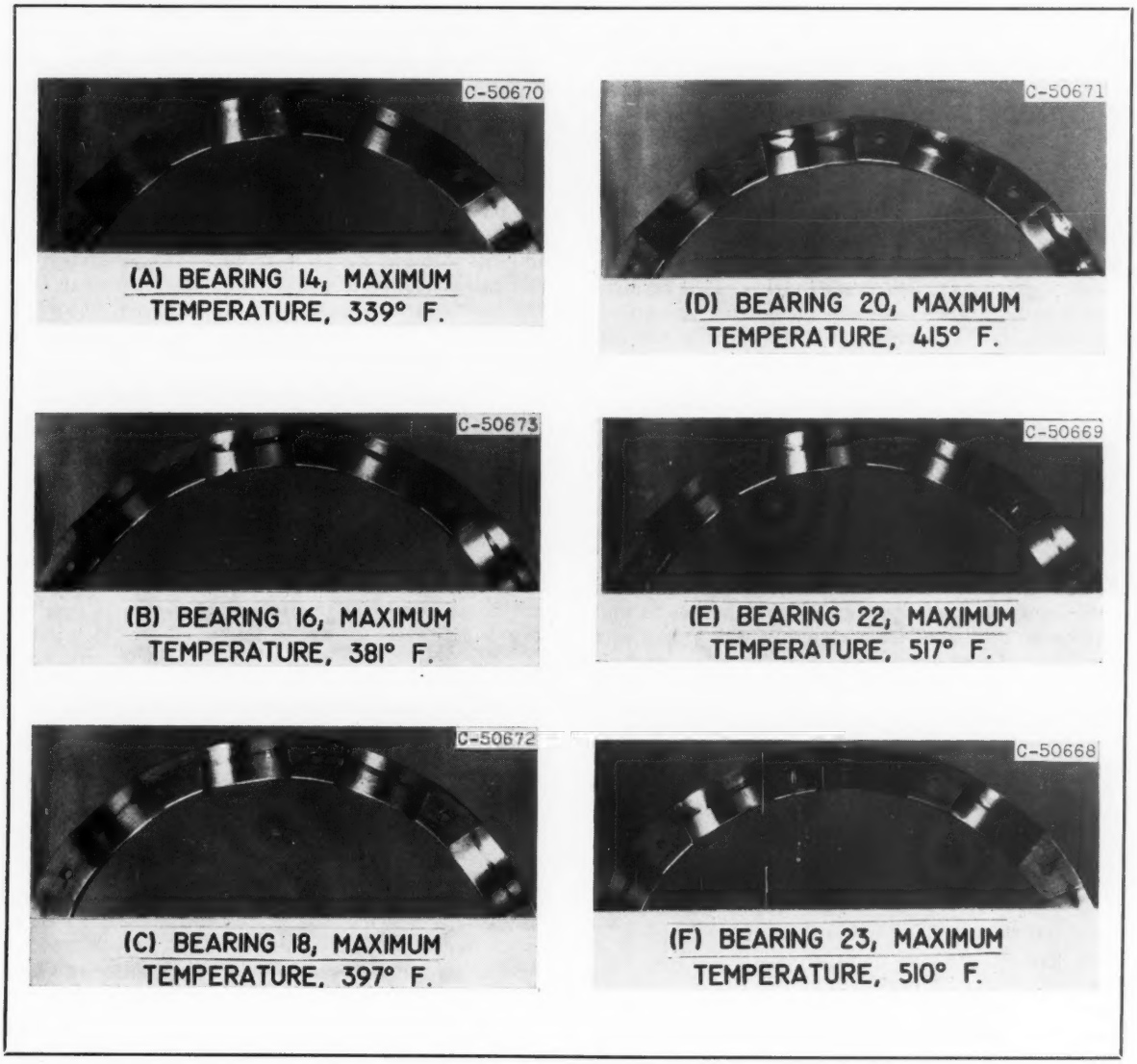
C-50670
(A) BEARING 14, MAXIMUM TEMPERATURE, 339° F.
C-50671
(D) BEARING 20, MAXIMUM TEMPERATURE, 415° F.
C-50673
(B) BEARING 16, MAXIMUM TEMPERATURE, 381° F.
C-50669
(E) BEARING 22, MAXIMUM TEMPERATURE, 517° F.
C-50672
(C) BEARING 18, MAXIMUM TEMPERATURE, 397° F.
C-50668
(F) BEARING 23, MAXIMUM TEMPERATURE, 510° F.

Fig. 12. Appearance of test bearing retainers showing ball pocket wear.

TABLE 1.—Resume of Minimum Oil Flow Test Bearing Data

BEARING NUMBER	TOTAL RUNNING TIME, HR	NUMBER OF INCIPIENT FAILURES	MAXIMUM LOAD, LB	MAXIMUM DN VALUE	MAXIMUM TEMPERATURE, °F	CONDITION OF BEARINGS AFTER TESTING
14	140	7	3000	12.0×10^5	339	No varnish deposits. Slight oxidation of unplated parts. No failure.
16	134	5	3000	10.5×10^5	381	No varnish deposits. Fatigue spall on I R.
18	122	13	3000	10.5×10^5	397	Slight varnish deposits. No failure.
20	168	16	1000	8.63×10^5	415	Slight varnish deposits. Extensive fatigue spalls on O.R.
22	238	22	3000	9.75×10^5	517	Varnish deposits on all bearing parts except balls. Balls heavily oxidized. No failure.
23	148	19	3000	10.5×10^5	510	Varnish deposits on all bearing parts except balls. Balls heavily oxidized. No failure.

Temperature influences the minimum oil requirements of a bearing partly through its effect on lubricant evaporation rate. At higher temperatures the lubricant evaporates faster so that a greater supply rate is required to maintain a lubricant film on the bearing parts. Thus minimum-oil-flow requirements increase at temperatures up to 400 F. At 500 F, however, required oil flows are less than those at 400 F at comparable DN values (Fig. 11). At temperatures above 400 F the synthetic lubricant used in these experiments begins to decompose at a rate great enough to deposit varnish on the bearing. This varnish appears to have some beneficial lubricating properties although it can be detrimental if the deposits are too heavy. Beneficial effects of the decomposition products of JP-4 fuel have been observed by Coit, Sorem, Armstrong and Converse (7). Carter (8) noted beneficial effects on ball fatigue life with MIL-L-7808 oil at 450 F, and, in unpublished NASA research with 20-millimeter-bore ball bearings, similar reductions in minimum-oil-flow requirements were noted at temperatures above 400 F.

Table 1 and Fig. 12 summarize qualitative data on lubricant deposits and wear of one bearing from each of the six bearing pairs tested. Bearing number 14 was tested at a maximum temperature of 339 F and was completely free of any varnish-like deposits. Bearing number 23 showed considerable varnish deposits after exposure to a maximum temperature of 510 F. Photographs of half the retainer from each of the six test bearings illustrating ball pocket wear are shown in Fig. 12. Varnish-like deposits, where present, have been removed with solvents. Ball pocket wear is heavy only in bearing 20 (Fig. 12[d]), which was subjected to a maximum temperature of 415 F. From these limited data, lubrication with this type of oil appears to be poorest in the temperature region immediately above 400 F. This is also the temperature region in which the required oil flows are highest (Fig. 11). The varnish deposits that occur at 500 F. are definitely beneficial in that they allow bearing operation at flows lower than those at 400 F and also help prevent wear. The magnitude of ball pocket wear in bearings 22 (Fig. 12[e]) and 23 (Fig. 12[f]) is considerably below that of bearing 20 (Fig. 12[d]).

There was no significant variation in the retainer locating surface wear in the six test bearings. An uneven wear due to retainer unbalance was observed in bearings 16 and 23. In the other test bearings, the retainer locating surface wear was not sufficient to break through the silver plate.

Two of the six test bearings sustained fatigue failures. The inner race of bearing number 16 fatigued, and tests of this bearing had to be suspended because of a change in bearing characteristics produced by the fatigue pit. The outer race of bearing number 20 sustained an extensive fatigue spall. These data are not sufficient to evaluate the fatigue performance of M-50 tool steel and MIL-L-7808 lubricant.

SUMMARY OF RESULTS

An investigation was conducted in an attempt to determine the minimum-oil-flow requirements of size 215 deep-groove, M-50 tool-steel ball bearings with silver-plated iron-silicon-bronze retainers. Tests were conducted at thrust loads of 1000 and 3000 pounds, DN values of 6×10^5 to 12×10^5, and temperatures to 500 F. Test bearings were air-oil mist lubricated with a MIL-L-7808 synthetic diester lubricant.

The results of this investigation are as follows:

1. Minimum-oil-flow requirements for continuous bearing operation increased with increasing DN value and also with increasing bearing load.

2. Bearings will operate satisfactorily at very low oil flows using an air-oil mist, once-through system of lubrication. As an example it was possible to run size 215 bearings satisfactorily at 12,000 rpm (9.0×10^5 DN), 400 F, and a 1000-pound-thrust load with oil flows as low as 0.003 pound per minute.

3. Minimum-oil-flow requirements increased with increasing bearing temperature up to 400 F. At 500 F varnish deposits on the bearing parts formed by the decomposing lubricant apparently aided lubrication and reduced the flow requirements below those at 400 F.

4. In the minimum-oil-flow range, bearing temperature and torque generally decreased as the oil flow was decreased. This is probably due to less heat generation through oil shearing.

5. An empirical relation which makes it possible to calculate the required minimum oil flow at a given DN and load is presented. This relation is considered to be valid only for the bearing size, range of speeds, and loads of this investigation.

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8. Carter, T. L.: "Effect of Temperature on Rolling-Contact Fatigue Life with Liquid and Dry Powder Lubricants." NACA TN 4163, 1958.

ASLE Technical Digest

Papers which, because of space limitations, cannot be published in full in Lubrication Engineering or ASLE Transactions.

"An Investigation of Oils for Use in Helicopter Oscillating Bearings," by L. A. Donnelly, Sikorsky Aircraft Division, United Aircraft Corp., Stratford, Connecticut

The lubrication of oscillating ball and roller bearings presents some special problems that are normally not significant in applications involving rotation without oscillation. The angular displacements encountered in the operation of helicopter rotor head bearings fall within the range which poses problems peculiar to oscillation.

A friction oxidation tester devised by the authors' company has been used to evaluate greases for oscillatory applications. The rig tests two tapered roller bearings mounted on a common shaft and each subjected to 5000 lbs. radial load and 4700 lbs. axial load. An oscillatory motion is applied to the system and the bearing drag is measured. Time to failure is taken as that time required for the drag to reach 60 lbs. For grease tests, good reproducibility is obtainable and satisfactory correlation exists between test rig life and bearing life in a rotor head.

Since oil lubrication of rotor head bearings is felt to provide certain advantages over grease lubrication, the test machine was modified to permit evaluation of oils. It was found that the degree of reproducibility with oil lubricants compared favorably with that obtained on similar types of lubricant testing apparatus. It was found that while the tester was capable of detecting differences between various oils, the oils as a group gave a much lower indicated life on the tester then the greases. This is contrary to experience with tests on rotor head units. In order to have a test that is capable of expressing the adequacy of both oils and greases in common units, it will be necessary either to change test speeds and loads in an attempt to discover one set of conditions that will rate oils and greases on a comparable basis, or else to find a relationship of "oil hours" to "grease hours" that will relate the test rig results to actual application experience.

Some tests were run to obtain insight into the mechanism of failure of the test bearings. Two separate and independent forms of failure were found to take place that can occur singly or together in a given test, depending on the lubricant. One type was a fatigue pitting that was always found to occur first at the small diameter of the cone and progress towards the large diameter. The other variety was surface damage of a fretting corrosion type that always occurred first at the large diameter of the cone and progressed towards the small diameter. The location of the areas of fatigue pitting and fretting corrosion can be shown to be consistent with the stresses existing in the cone.

Silicone oils were found to mainly produce the fretting type of damage, whereas petroleum and synthetic base oils exhibited the pitting type of failure.
(Digested by H. J. Connors)

"Additive Studies in Research on High Temperature Gas Turbine Lubricants," by H. W. Adams, Wright Air Development Division, Wright Patterson Air Force Base, Ohio

One of the broad problems of military aircraft lubrication is the development of lubricants with wide fluid ranges capable of operating in severe thermal and oxidative environments. The lubricant for an advanced gas turbine engine, for example, must perform satisfactorily at bulk oil temperatures of 425 to 450F with hot spot temperatures of 650 to 700F and still have reasonable pumpability properties at —65F.

To help meet the need for effective lubricants, a program was carried out to develop an engine oil for operation at 400 to 500F using 10, 10 dipheny phenozasiline, commonly referred to as 5-10-10, as an antioxidant.

Considerable oxidation study on 5-10-10 indicated that specific thermal levels were necessary to activate it as an antioxidant. Such studies led to combining 5-10-10 with other well known lower temperature antioxidants to form an additive package. The results of this work showed that phenyl alphanaphthylamine (PANA) was the best commonly available antioxidant for this purpose. The additive package was found to be particularly effective in trimethylol propane esters.

Tests conducted on a large scale roller bearing rig and on an actual aircraft gas turbine engine showed the additive package to be more effective then the more common high temperature formulations in the 425F range. A lubricant formulation using 5-10-10 and PANA as an additive package was run for a full 100 hour test in a gas turbine engine at 425F. The viscosity change was minute and the acid build up value was very small. Dissembly of the engine revealed it to be very clean with little coking on the components.
(Digested by H. Connors)

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Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

A. D. Newman (Pametrada), "Water-Lubricated Bearings for Marine Use". *Journal of the American Society of Naval Engineers, Inc.*, v. 70, no. 4, Nov. 1958, pp. 756-760.

In the design of water lubricated bearing systems, viscosity, density, and specific heat are the principle properties of water that should be considered. As with oil there are two important regimes of bearing operation—hydrodynamic and non-hydrodynamic lubrication. For non-hydrodynamic lubrication, it is necessary to choose bearing materials that are compatible with water as the lubricant. Test results show that a fabric-reinforced phenolic resin and an asbestos - reinforced phenolic resin operate at loads up to 500 psi and a surface speed of .9 ft/sec with reasonable values of friction and wear. Hydrodynamic lubrication with water does not differ significantly from that with other incompressible fluids. The low viscosity of water and the consequent small film thickness must be considered when designing the bearing system. Increased attention must be given to conditions such as misalignment, corrosion and dirt. A properly designed water lubricated bearing will operate with lower power loss and less temperature rise which are particular advantages for high speed applications. As a result of the high cooling capacity of water, low-speed bearing applications such as stern tube bearings can accept high loads and thus allow shorter bearing lengths. Other advantages of water lubrication are simpler sealing arrangements and elimination of fire hazards inherent in an oil system. In particular applications, water lubricated bearings have been successfully used in high speed turbines, auxiliary machinery and slow moving, high load bearings. *(Abstractor: H. Apkarian)*

W. L. R. Rice, D. A. Kirk, W. B. Cheney, Jr., "Radiation-Resistant Fluids and Lubricants," *Nucleonics* v. 18, no. 2, 1960, 67-71.

A Summary of some results of a test program conducted at Materials Laboratory, Wright Air Development Center, and review of present state of art. Describes several solid film lubricants, greases and fluids showing promising radiation resistance and high temperature stability. Of particular interest as base fluids are polyphenyl ethers, which can stand 800° — 900 F and 10^{11} ergs-cm. *(Engineering Index Abstract, per W. E. Campbell)*

L. D. La Croix, M. L. Kalinowski, "One Test Evaluates Motor Oil Contribution to Combustion Deposits." SAE Paper n. 126 U, October 1959, 22 pp. also SAE Journal, v. 68, no. 3, March 1960, pp. 94.

Described is a laboratory engine test to determine the Octane Requirement Increase (ORI) due to combustion chamber deposits in high compression engines, as well as the effect of deposits on rumble, surface ignition, and spark plug fouling. The data shown compare seven 10W-30 motor oils and their effect on engine performance. While the base oil and several elements did slightly affect engine operation, the effect of .19% phosphorus was outstanding. A difference in the ORI after deposits formed was found to be 11.5 Performance Numbers and the surface ignition octane requirement was reduced 4 Numbers with oil containing .19% phosphorus compared to the base oil. Of interest is that no noticeable improvement occurred in surface ignition or rumble up to a phosphorus content of about .1%. Spark plug life was increased up to 240% of that with base oil, but as little as .05% phosphorus would increase plug life to 170%. Some data are shown for 12:1 and 8.0:1 C.R. engines, but most results are for a standard 9.5:1 production V-8. The appendix describes the methods of measuring surface ignition, rumble, plug fouling and ORI, and the oils and test fuel used. *(Abstractor: George C. Lawrason)*

"Lubricant Behaviour in Concentrated Contact Systems—The Castor Oil-Steel System." F. W. Smith (Nat. Research Council of Canada, Ottawa), Wear-Usure-Verschleiss, v. 2, no. 4, May, 1959, 250-63.

Experiments are described in which frictional forces are measured between hardened steel surfaces lubricated with castor oil. In one series, balls of different sizes slide against a moving plane at a maximum Hertz stress of 389000 psi (273.5 kg/mm²) over a range of speeds up to 140 cm/sec. In another experiment the frictional force is measured between rollers which roll with a small component of sliding motion, the elastic stresses being similar to those used in the first experiment. It is concluded that in a certain range of speed most of the frictional force is transmitted through a film of lubricant acting as a plastic solid whose shear strength decreases with increasing temperature, and that in the sliding experiment the coefficient of friction gives essentially a measure of the temperature rise in the lubricant. It is further suggested that in concentrated contact systems of this type where the frictional force is determined by the high-stress plastic yield of the lubricant, the film thickness is determined by viscous flow properties. *(Author's summary: per E. A. Ryder)*

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Patent Abstracts

Compiled by Ann Burchick
Aluminum Co. of America

Mold Dressing, Patent No. 2,956,890 (J. M. Casteras, assignor to International Smelting and Refining Co.) A mold dressing for coating the molding surface of a mold for molten metal consisting essentially of a light hydrocarbon oil which contains approximately 0.1 to 2.0 percent by weight of a lecithin and in which there is dispersed approximately 1 to 10 pounds of a finely comminuted carbon per gallon of oil.

Lubricating Oil Compositions, Patent No. 2,944,974 (L. E. Lorensen, J. Zachar and R. C. Jones, assignors to Shell Oil Co.) A mineral lubricating oil composition comprising a major amount of mineral lubricating oil and a minor amount, sufficient to improve the pour point, viscosity index and detergency of the oil, of an oil-soluble copolymer of (1) a monovinyl-substituted pyridine of the group consisting of an unsubstituted vinylpyridine and derivatives of the said vinylpyridine having a lower alkyl group substituted on a ring carbon atom, (2) a mixture of a stearyl ester of an acrylic acid of the group consisting of acrylic acid and methacrylic acid and a lauryl ester of an acrylic acid of the group consisting of acrylic acid and methacrylic acid in the mol ratio varying from 1:4 to 4:1, respectively, and (3) a C_{1-4} alkyl ester of an acrylic acid of the group consisting of acrylic acid and methacrylic acid in an amount of from 10 to 60 mol percent of the ester mixture of (2), the mol ratio of (1) and the ester mixture of (2) and (3) varying from 1:2 to 1:10, respectively, said polymer having an average alkyl chain length of from C_{12} to C_{14} carbon atoms, the copolymer having a molecular weight of from about 5×10^4 to 2.5×10^6.

Stable Anti-Corrosive Lubricant Composition, Patent No. 2,944,971 (M. P. Kleinholz, assignor to Sinclair Refining Co.) A lubricating oil composition consisting essentially of a major proportion of mineral lubricating oil and minor amounts of an oil-soluble nickel dithiophosphate diester in which each organic group contains from about 4 to 14 carbon atoms, and a basic barium mahogany sulfonate formed by neutralizing a mahogany sulfonic acid with at least about 1.5 equivalents of a basic barium compound and wherein the barium content is from about 0.2 to 2.0 percent by weight and the phosphorus content is from about 0.03 to 0.3 percent by weight based on the total weight of said composition.

Water Base Lubricant Containing Dimethyl Sulfoxide, Patent No. 2,956,951 (M. J. Furey, assignor to Esso Research and Engineering Co.) A water base lubricant composition comprising a major proportion of water, 1 to 5 wt. percent of a water soluble ether cellulose thickener and 1 to 3 wt. percent of a dialkyl sulfoxide having in the range of 1 to 3 carbon atoms in each alkyl group, wherein said lubricant composition is a clear solution.

High Temperature Grease Compositions Containing Salicylic Acid Derivatives, Patent No. 2,944,970 (W. H. Peterson, assignor to Shell Oil Co.) A high temperature grease composition consisting essentially of a major proportion of a mineral lubricating oil, a gelling proportion of a non-soap colloidal gel of inorganic origin of the group consisting of amorphous inorganic oxides, clays and onium clays, and from about 1% to about 10% by weight of a hydroxy carboxy aromatic substance.

Marine Diesel Lubricant, Patent No. 2,944,-021 (C. J. Ulzheimer and J. C. Zimmer, assignors to Socony Mobil Oil Co., Inc.) An emulsion lubricant for marine diesels which consists of, by weight of the lubricant, between about 10 percent and about 35 percent water, between about 80 percent and about 55 percent mineral lubricating oil having a Saybolt Universal viscosity of between about 75 seconds and about 80 seconds measured at 210 F., between about 1 percent and about 5 percent alkali-metal hydroxide, said alkali-metal hydroxide being present in the water phase of said lubricant, the amount thereof being sufficient to effect a total base number in the lubricant of between about 10 and about 50, between about 1 percent and about 10 percent oil soluble polyvalent metal sulfonate and between about 3 percent and about 10 percent esters of high molecular weight acids obtained by controlled oxidation of petroleum oil.

Di-Ester Fluids with Improved Water Tolerance, Patent No. 2,944,973 (T. W. Langer and B. F. Mago, assignors to Union Carbide Corp.) A lubricant and hydraulic fluid having improved viscometric stability and low temperature water tolerance, consisting essentially of a mixture composed of at least one aliphatic di-ester selected from the group consisting of the higher alkyl adipates, sebacates, azelates and pimelates, and from 2 to 22½ percent of at least one high viscosity alkyl polyoxyalkylene glycol ether selected from the group consisting of lower alkyl mono- and di-ethers of polyoxypropylene and polyoxyethylene glycols, the viscosity of said ether ranging from about 59 Saybolt Universal seconds to about 3000 Saybolt Universal seconds at 100 F.

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New York Section Chairman, S. E. Manilych (left), is shown with Charlie Bailey, Lubrication Engineer, National Tube Division, United States Steel Corporation. Mr. Bailey presented "Frontiers in Plant Lubrication" at the May, 1961 meeting of the New York Section. The United States Steel Corporation's Mobile Training Unit was also on exhibit.

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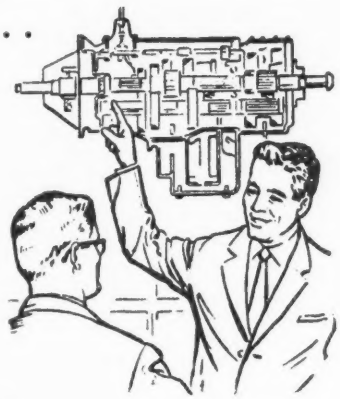

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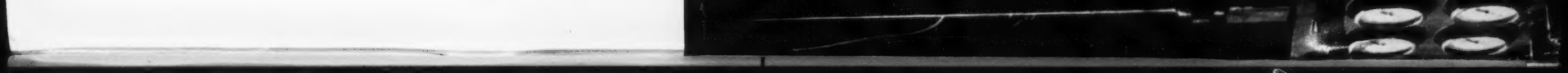

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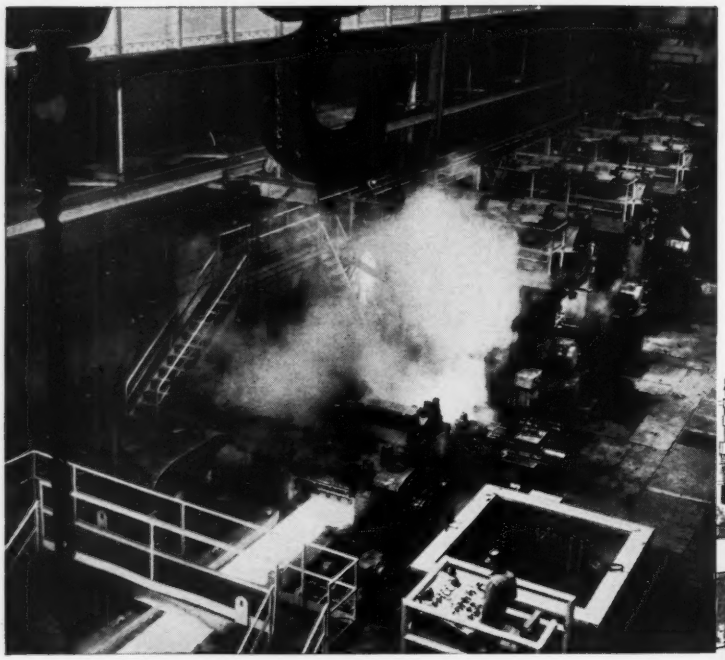

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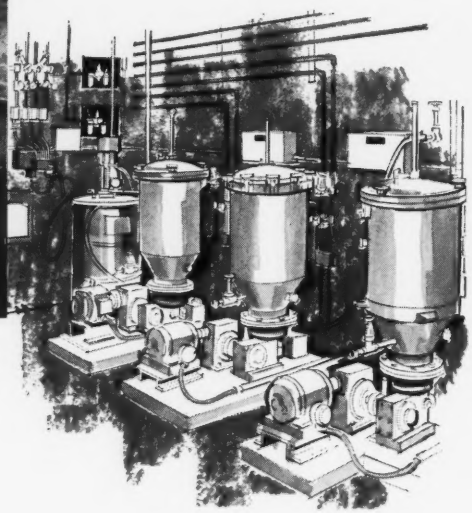

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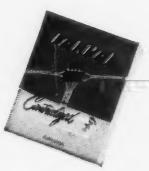

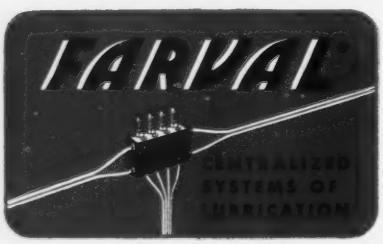

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